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Design and Construction of the Polematcher, Main Frame, and Transmission Drive for Wind Furnace I (WF-1)

Item Type	article;article
Authors	Antoon, Fred A.
Download date	2026-06-09 20:59:36
Link to Item	https://hdl.handle.net/20.500.14394/50472

UNIVERSITY OF MASSACHUSETTS/AMHERST
ENERGY ALTERNATIVES PROGRAM



DESIGN AND CONSTRUCTION OF THE
POLEMATCHER, MAIN FRAME, AND
TRANSMISSION DRIVE FOR WIND FURNACE I (WF-1)

A Report Presented

by

Fred A. Antoon

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Energy Alternatives Program
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Amherst, Massachusetts 01003

August 1978

WF/TR/78/14

ACKNOWLEDGEMENTS

I would like to express sincere gratitude to my committee members, Professors Duane E. Cromack, William E. Heronemus, and Armand J. Costa. Because of their enlightenment, inspiration, and hard work, I will be forever dedicated to the development of alternative energy sources.

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I. INTRODUCTION

On November 1, 1976, a wind machine of character much different than those in existence today was erected and put into service in the Orchard Hill area of the University of Massachusetts, Amherst, Massachusetts. The machine is a horizontal axis, high speed, 3-bladed propeller type with continuous pitching blades. The design rated conditions are to generate 25 kilowatts of electricity in a 26.1 mile per hour wind speed. All of the engineering and construction of this wind turbine was done at the University of Massachusetts, although in some areas, outside conceptual ideas were used. For this wind turbine, one of the most important design and construction parameters was cost.

In light of today's energy concerns, much interest in alternative sources of energy has been sparked. At the present time, industry does not provide an economic wind machine. This must change soon because, with time, the cost of our usual energy sources continue to rise. We are at a point now where depletion dates can be fairly accurately predicted for some of our energy sources. It has been this concern for the running out of certain energy forms that has led to the investigation of the extraction of continuous renewable energy sources. Windpower, as a renewable energy source, is considered here.

In order to develop an economic wind turbine, technological brainstorming and experimenting are necessary. It has been the purpose and labor of this project to begin such work.

Specifically, this project deals with the design and construction stages of three major components of the wind turbine. They are the pole-matcher, main frame, and power transmission components. An important aspect

of this project report is the incorporation and in some cases, identifying of tradeoffs and methods used to construct an economic, energy efficient wind turbine. Also included are comments on how the final methods and designs could be improved, since experience being a good teacher, certain designs were realized too late to be incorporated into the existing wind turbine.

II. BASIC COMPONENTS AND SPECIFICATIONS

The design and construction criteria of the UMass wind turbine has two major considerations in mind. One is that the machine can turn an electric generator within itself sending energy to the ground through transmission wires; the other is to supply shaft work at the ground level. The former consideration has been labeled Wind Furnace-I, (WF-1), and the latter, Wind Furnace-4 (WF-4).

A. Wind Furnace I (WF-1)

WF-1 is currently that WECS which operates on the UMass Campus. The main components, to be considered in this report, which make up that system are the polematcher, main frame, and the truck rear axle assembly.

The polematcher is a structural member which supports all of the wind turbine's loads and transfers them into the tower. The weight and reaction to the energy being developed by the rotor pass through an attached tapered roller bearing about which the machine is free to yaw. There is a second plastic radial bearing approximately 12" below which absorbs the second reaction load. Between these two bearings is a set of sliprings which are used to transfer the energy from the generator across the yaw axis and into the ground transmission lines.

The main frame is the structural beam about which the wind machine is built. Its function is to transfer wind and weight loads of all its attached components to the polematcher. The frame supports a 25-kilowatt generator (rated at 1800 RPM) on one end and a wind rotor (rated at 167 RPM) on the other end. The placement of the generator and wind rotor are such that the main

frame balances at the polematcher. Because of the difference between the wind rotor and the generator rated RPM, a speed drive with a step-up ratio of 1800/167 or 10.82:1 is required.

The truck rear axle assembly has a multi-purpose. It helps to stiffen the main frame in that it is built to support heavy loads. It has a specially treated axle shaft and gears to transmit high torque requirements, a drum brake to slow or stop the wind turbine, and a rotating hub to which the wind rotor is attached. The hypoid gears, which are encased in the differential housing, comprise the first of a two-stage step-up drive. They are a 90° right angle drive and have an overall ratio of 4.86 to one. For the WF-1, those gears are angled to the side so that a chain and sprocket second stage speed drive could be added. If shaft power is desired at ground level, then these gears plus the whole differential axle assembly would be rotated 90° until the differential pinion shaft points straight down the tower. Then with a short universal shaft arrangement connecting the axle drive through the polematcher to the vertical shaft, WF-4 would be created.

One last component that should be mentioned here is the second stage speed drive. Considering that the hypoid gear drive has a 4.86 to one ratio, then in order to achieve the overall desired drive ratio of 10.82 to one, the second stage speed drive must have a ratio of 2.22 to one. The second stage transmission is built of 3" wide silent chain on 42" centers. The chain is lubricated in an oil bath consisting of "Type A" or "Dexeron" automatic transmission fluid.

B. Wind Furnace 4 (WF-4)

WF-4 is a wind system which does not utilize a wind generator aloft, but instead drives a mechanical churn. Once the generator is removed though, ballast should be added in its place in order to maintain the wind machine balance. Instead of electric, shaft power is brought from the wind turbine through bulkhead bearings located within the tower and then to the ground where it would be used in whatever form is desired. A yaw driver may now become necessary so that the reaction from the vertical shaft torque does not yaw the machine from the wind direction. Sliprings for this model may not be necessary at all.

C. Summary

The conceptual arrangements of WF-1 and WF-4 had long been laid out before the actual identifying of specific designs and materials. The primary wind turbine designs were begun after the basic elements of the machine were selected. The most important element of the wind turbine about which all designs were centered was the truck rear axle assembly. As discussed later in this report, much consideration was given to choosing this rear axle, the most important being cost versus whether it could be modified for its new role. The next element was the generator and finally sprockets and chain as the final link in the drive transmission. With these elements in mind, the primary designs were begun.

III. PRIMARY DESIGNS

Before a realistic design could begin the question of whether to build structural components out of laminated fiberglass or steel had to be evaluated. Due mainly to the fact that time was hurried and this designer knew little of working the fiberglass laminates, the steel design was chosen. Some interesting points in design of fiberglass laminates were raised and presented here.

A. Fiberglass Laminated Design

Probably the greatest advantage to building a fiberglass design is that expensive machines and machinery are not required. The fiberglass components could be molded in pieces and then bonded together with fiberglass and resins or one complete section could be done at one time. The man-hours required for these jobs decrease as experience and knowledge increase. This was demonstrated during the UMass windpower project while constructing the wind turbine's blades. The basic design entails molding a supporting platform with ribbed sections emanating from the tower or pole-matcher section. The truck axle would not be permanently embedded into the fiberglass, but it could be used along with a releasing agent to have its form permanently molded into a fiberglass bed. That way the truck axle could be easily taken out for repair. Mounted to that platform would be the various other accessories such as a second speed drive, generator, and batteries. A channel above the polematcher would still be provided in the event that shaft power at the ground level is desired. An initial disadvantage is the cost of building the molds, but this cost would be returned

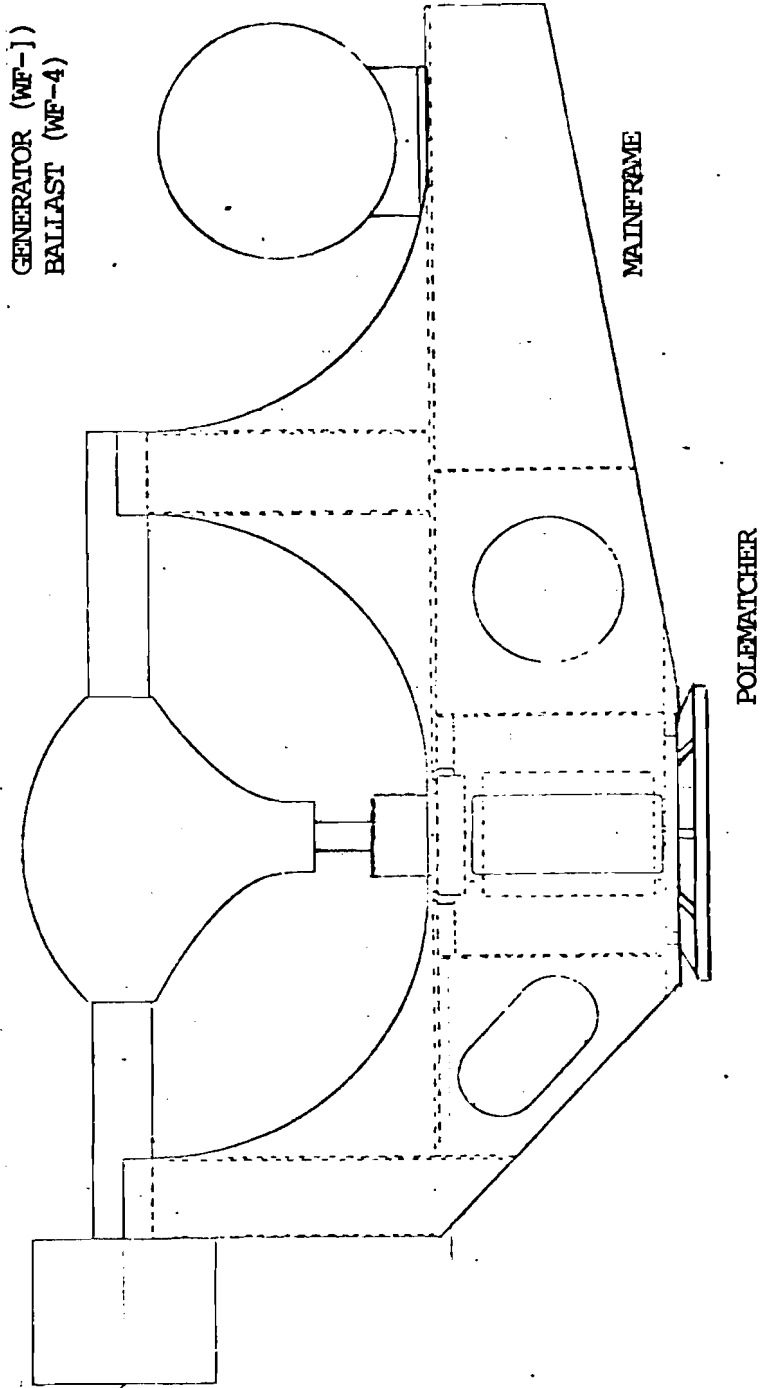
as more machines were turned out. The advantages would be a structure of low weight, high strength, and no machining, cutting, or welding costs.

B. Steel Design

The steel design was thought out and constructed from the point of view that steel would be bought in the forms of tubing, flat plate and angle iron. The procedure used from the initial idea to the finished product is as follows. A conceptual idea for the entire machine was formed after reading material on past windmill designs and after engaging in discussions with other engineers on the project. A working design was sketched out and discussed again. After more refinement, the design was presented at a regular windpower group meeting where other criticisms were voiced. During this time, two $\frac{1}{2}$ scale cardboard models were built, the second being a revision of the first. The models clearly showed how all systems interacted, that is specific placement of the main frame polematcher generator and truck axle.

With the acceptance of the final layout, individual assemblies and subassemblies were identified. Specific designs for each were carried out and evaluated in the same manner as the overall design with the exception that the machine shop engineer, Professor Armand Costa guided the designs for efficient machining. In many ways the advice of this man led to good engineering practice in the final design. Figure 1 shows the accepted concept of the wind turbine with the relative positions of the basic elements. In the Figure, the differential pinion shaft is angled toward the ground for the WF-4 configuration. Component "A" would be ballast weight.

TRUCK REAR AXLE



GENERATOR (WF-1)
BALLAST (WF-4)

MAINFRAME

POLEMAICHER

WINDFURNACE CONCEPTUAL LAYOUT
FIGURE 1

For the WF-1 configuration, the differential pinion shaft is rotated up 90°, Component "A" is the generator and a sprocket and chain drive is added. This model is shown in the wind turbine's final assembly drawing later in this report.

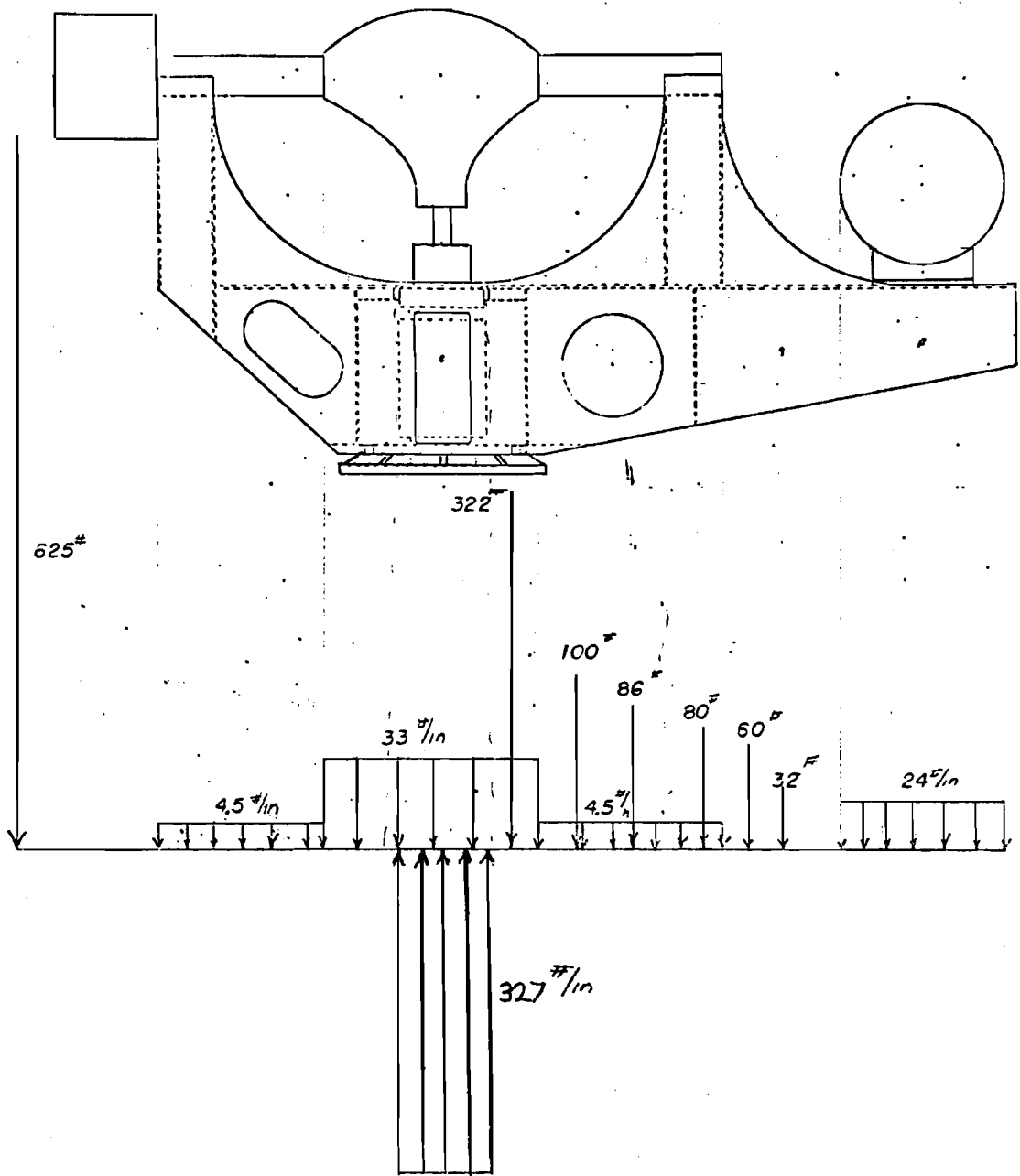
IV. STRUCTURAL LOADS

In order to determine accurately the stresses to which the machine would be subjected, a good evaluation of weight and wind loads had to be done. In the analysis, only static and no dynamic forces were considered. Only after construction began were the dynamic loads assessed. In most instances the static evaluation was sufficient. Only one dynamic load was of any real concern. That was a gyroscopic couple induced by the simultaneous occurrence of rotor and yaw spin rate. This was finally controlled by a yaw damper that limited the maximum yaw rate that the wind could induce on the machine.

A. Weight Loads

When the analysis was first begun, initial estimates for each component was made. The estimated weight of all supported components less the nacelle's main frame, lightning rod, and yaw damper, was 1650 pounds, the actual weight was 1590 pounds. Once all the extras were added the total combined weight was 2535 pounds.

Figure 2 shows the current weight distribution identifying most components as point loads. In comparison to this loadings moment diagram, the current estimate compares well with the first attempt from which the original design stemmed. An unusual loading condition which is of concern occurs when the machine is mounted to the tower before the tower is erected. The machine is now lying on its side and so the loading condition now changes to 90° of the normal position. The loading force will increase as the tower is lifted into its operating position. From the weight and balance table in Appendix I, the vertical centroid is 9.76 inches above the top of the polematcher or 13.3 inches above the main bearing.



APPROXIMATE MAINFRAME LOADING

FIGURE 2

Knowing the weight of the machine and using a simple beam calculation, the radial force on the main bearing and polematcher is 5070[#]. If the machine is initially accelerated from the ground at 1.5 times the acceleration of gravity, then this force becomes 7605[#], which becomes a design criterion.

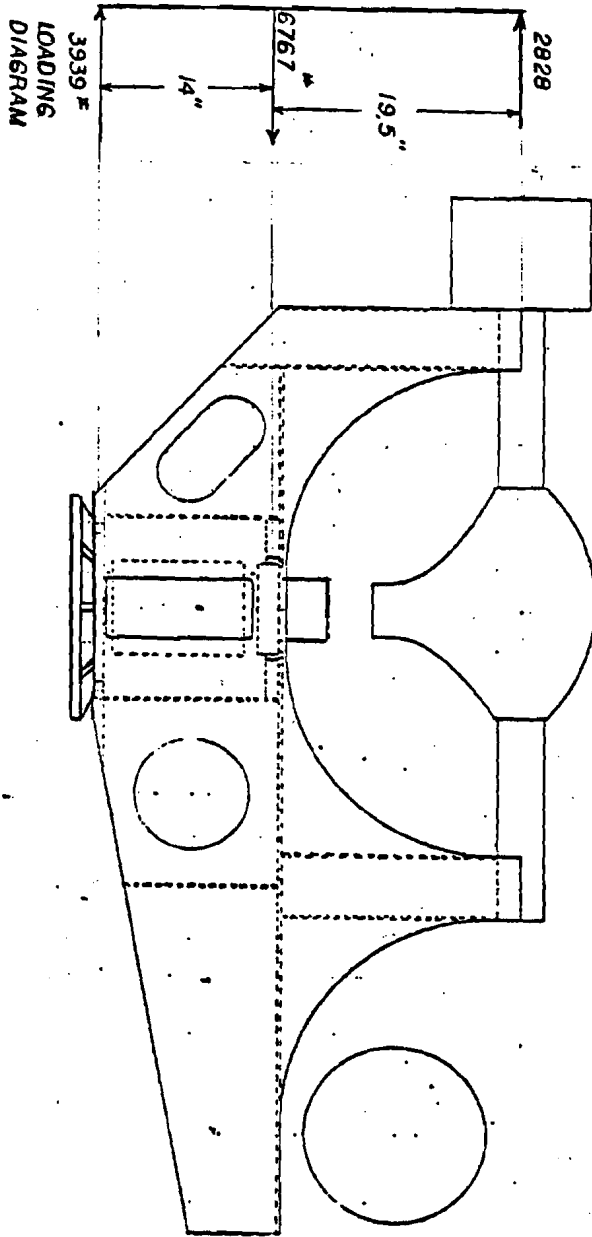
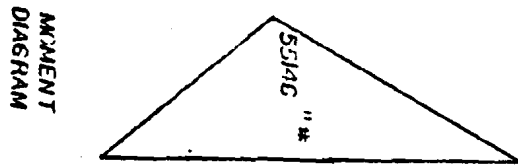
B. Rated Loads

The rated machine loads are determined mainly by the power that the machine is designed to extract from the wind, the wind-speed that corresponds to this power level, and the designed ratio of blade tip speed to windspeed or $\frac{\Omega R}{V_0} = 7.45$, a design power rating of 25 kilowatts and a design windspeed of 26.1 mph. Using the blade radius of 16.25; design $\frac{\Omega R}{V_0}$, and the rated windspeed of 26.1 mph, the rated shaft speed becomes 167 rpm. If a 67% drive efficiency is assumed, then 37 Kw is required at the windshaft, and for 167 rpm, the corresponding torque would be 1567 ft-lbs. Here an assumption was made that although an overspeed would increase the shaft torque vibrational transients or resonant frequencies would not. A report published in the UMass wind research progress report for the period May-December 1976 supports this assumption.

Wind rotor thrust is calculated from the ideal theoretical thrust - $4/9 \frac{1}{2} \rho A V_0^2$. For this machine at 26.1 mph, the thrust is 642 pounds adding 10% for nacelle drag is 707 pounds. Realizing that at rated windspeed the probability of a fast-acting gust is high, the design thrust force was chosen as four times the ideal thrust for 26.1 mph. Because thrust is proportional to V_0^2 , than a factor of 4 corresponds to $(4)^{\frac{1}{2}}$ or two times the rated windspeed. The

design thrust becomes 2828 pounds, Figure 3.

One other rated load was the moment of angular momentum induced at the wind rotor due to simultaneous yaw and rotor spin rate. In a UMass wind energy report, "Design of Hub and Pitching System for WF-1", June 1977, it was shown that a moment of 400 ft-lbs would exist if the yaw rate were allowed a maximum of one rpm. A yaw damper was added to control this. Hurricane force winds on a shutdown machine using nacelle drag only were two magnitudes of order smaller than the thrust values for doubled windspeed, therefore, hurricane winds pose no threat so long as the machine is shut down.



RATED THRUST LOADING

FIGURE 3

V. MAIN COMPONENTS

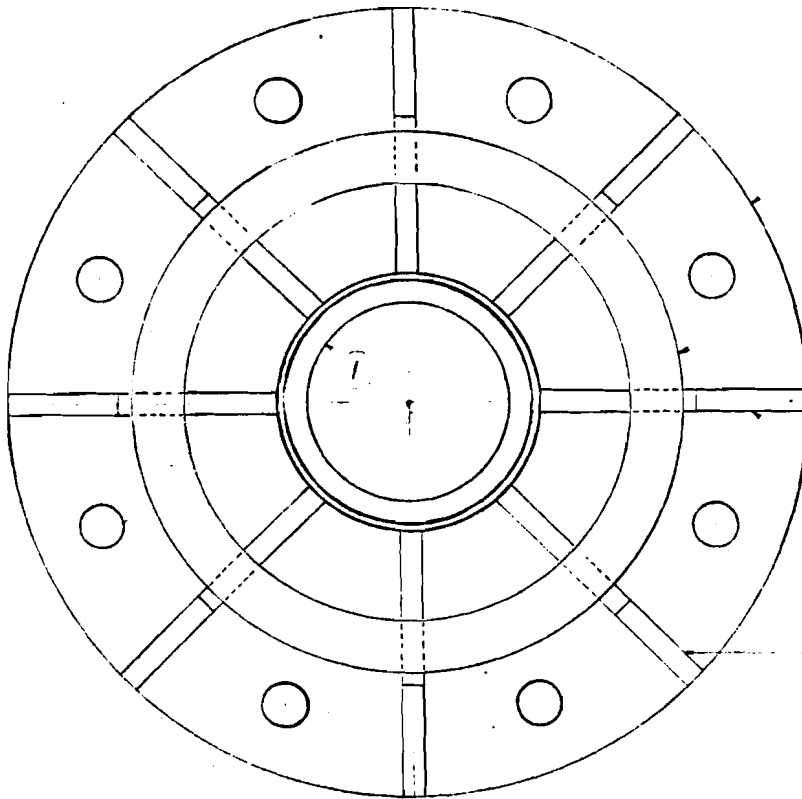
The main components of the WF-1, which this report deals with are polematcher, main frame, and transmission drives. In the following section the design construction and comment on each element is discussed. From the previous section, loadings were identified with probable estimates of transient or unusual forces. Factors of safety will be applied in this section and refer to the point of permanent yield. Each component described is that specific one used in the existing WF-1.

A. Polematcher

1. Design

Figure 4 is the completed polematcher less the slipping stack. Individual parts drawings appear in Appendix IV. The main element of the polematcher is a 19" standpipe fastened to a baseplate and eight $\frac{1}{2}$ " ribs. Fastened to these ribs is a pitch bearing support ring positioned high enough above the baseplate to allow the placement and tightening of the main tower to the polematcher bolts.

Figures 5 and 6 show the polematcher loadings described previously. The main structure is a $5 \frac{9}{16}$ " OD pipe with a $\frac{1}{2}$ " wall thickness and a section modulus of 9.25 in^3 . The figure identifies three loadings and two points "A" and "B". These loads are the reactions to the wind and weight forces acting on the main and lower skirt bearings. At points "A" and "B" elements of stress are evaluated using a Mohr's circle approach. The stresses which result are those due to shear and the combined bending and direct stress. Maximum shear stress for a relatively thin-walled tube is given as $2 \frac{F1}{A_p}$



2

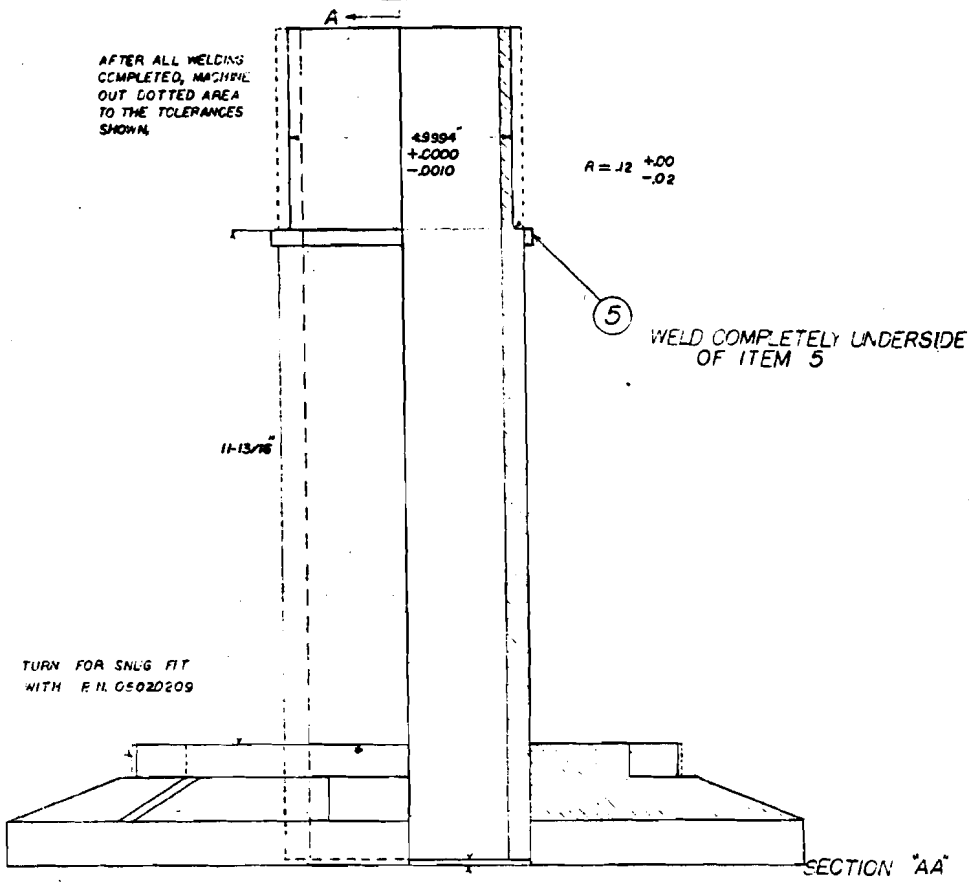
3

A

4

ELFT RIBS CENTERED EVERY
22.50 DEGREE FROM THE
BOLT HOLE CENTERS.
WELD COMPLETELY ALONG THEIR
BOACERS WITH ITEMS 1, 2 & 3.

AFTER ALL WELDING
COMPLETED, MACHINE
OUT DOTTED AREA
TO THE TOLERANCES
SHOWN.



$R = .12 \begin{matrix} +.00 \\ -.02 \end{matrix}$

5

WELD COMPLETELY UNDERSIDE
OF ITEM 5

1 1/16"

TURN FOR SNUG FIT
WITH F.H. 05020209

SECTION "AA"

TOLERANCES UNLESS
OTHERWISE SPECIFIED

±.01

SCALE 1/2

1.25"

FIGURE 4

ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
FLEXIBLE ENERGY - SUB ASSEMBLY	
DATE	12 NOVEMBER 1975
DRAWN BY	05020210

occurring at the neutral axis. Bending stress is M/S where "M" is the moment at the point considered due to F_1 and "S" is the section modulus. Direct stress is taken as F_3/A_p . Using a Mohr's circle, the maximum principle stress for the compressive load which here is greater than the tension load is:

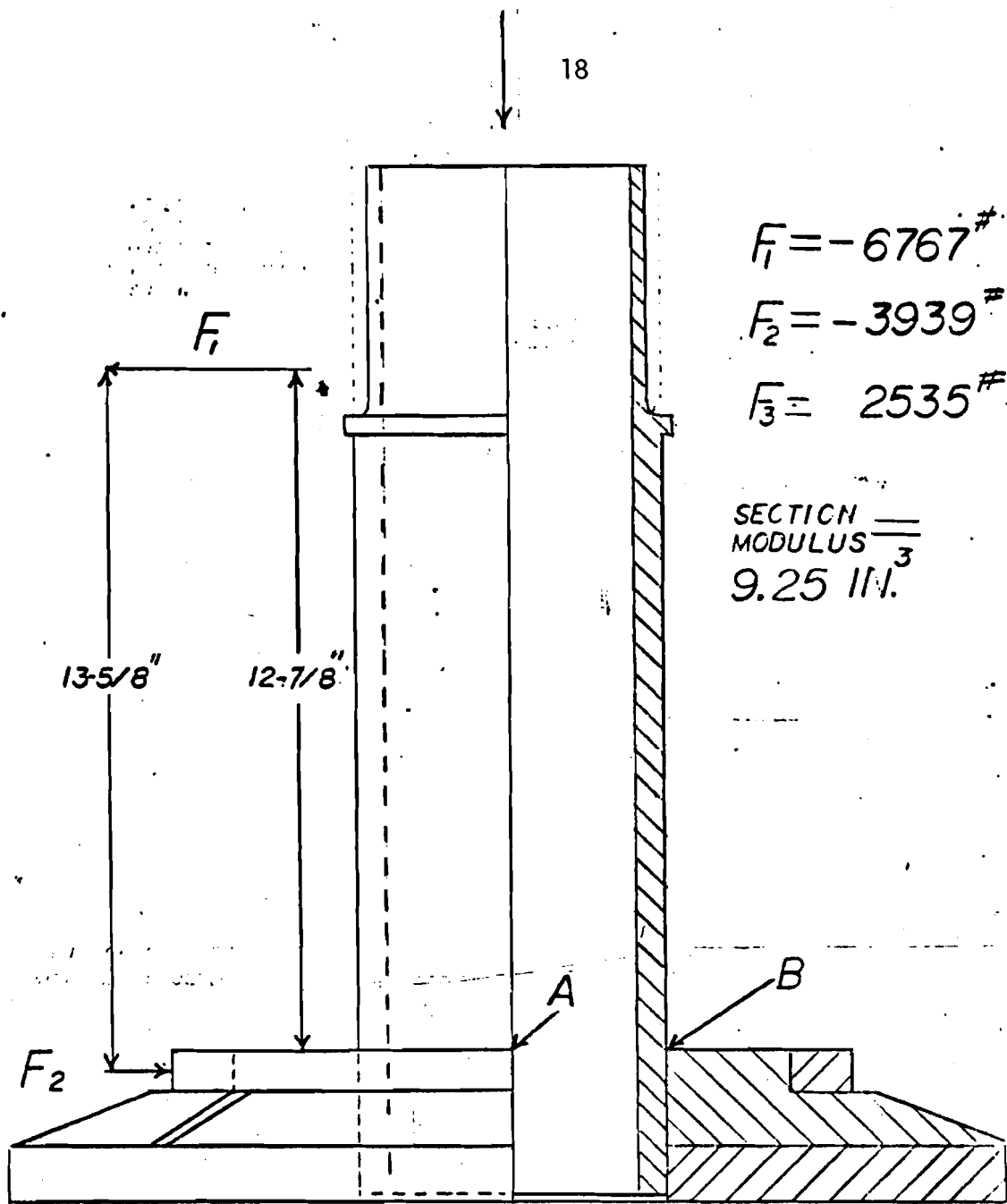
$$-\frac{(F_3/A_p + M/S)}{2} - \left[\left[\frac{F_3/A_p + M/S}{2} \right]^2 + \left[\frac{2F_1}{A_p} \right]^2 \right]^{1/2}$$

The maximum shear stress is:

$$\left[\left[\frac{F_3/A_p + M/S}{2} \right]^2 + \left[\frac{2F_1}{A_p} \right]^2 \right]^{1/2}$$

It is these values for stress which appear in Figures 5 and 6 for their respective load cases F_1 , F_2 , and F_3 . After this analysis it was decided that this tubing would be most adequate. Its availability was checked from a local distributor and a quote for the material requested. Placement, number and dimensions of the polematcher ribs was discussed with a professor of civil engineering whose analysis entailed a simple beam calculation for an otherwise complex stress problem. In the simplified analysis, one rib is allowed to support the entire load, with this a safety factor of three is achieved. After the polematcher was assembled, it was ready to be dressed with sliprings.

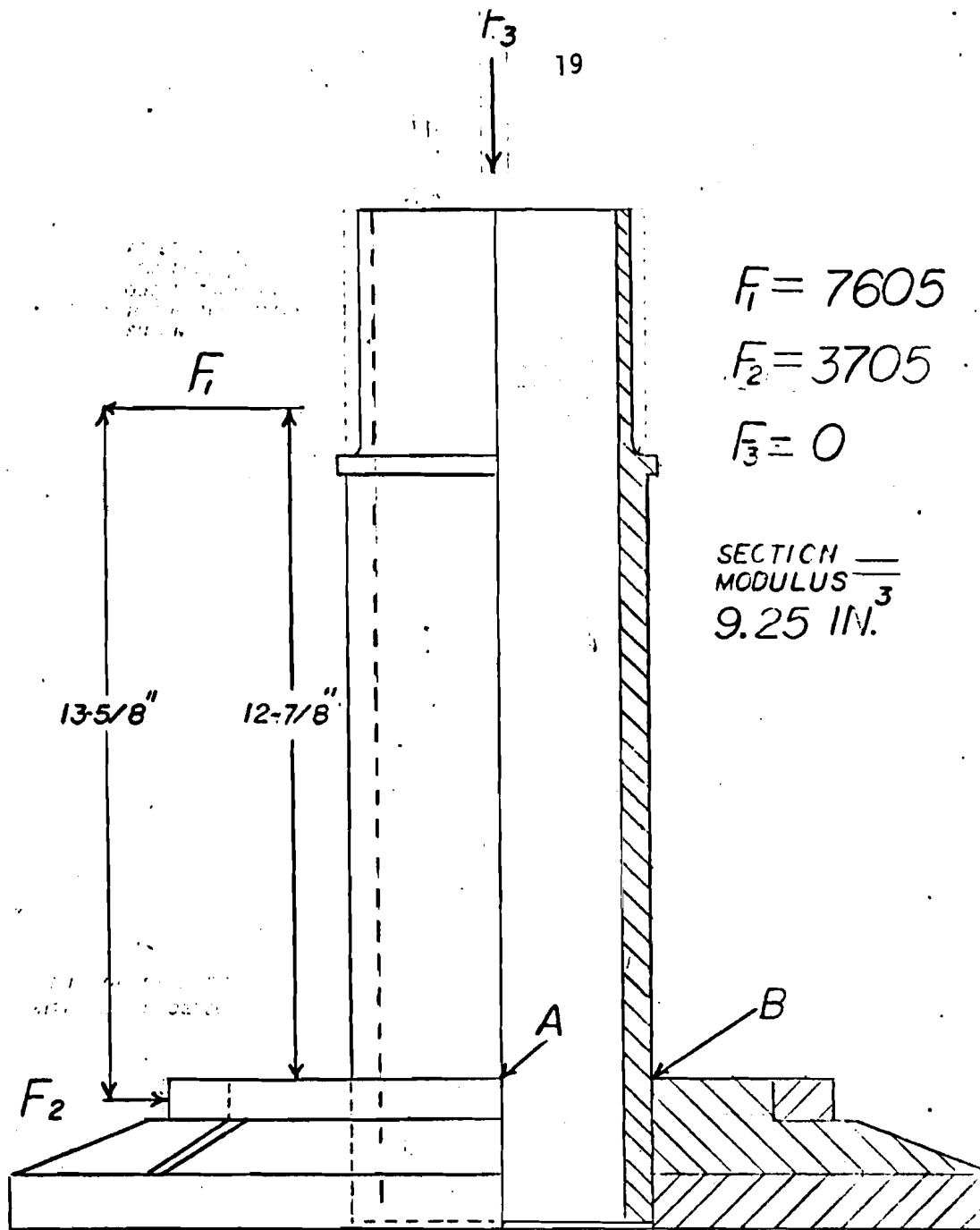
To decide on the actual number and type of sliprings, meetings with the electrical engineer in charge were held and



	A	B
PRINCIPAL STRESS PSI.	-1869	- 9738
MAXIMUM SHEAR PSI.	1709	0
MIN. S. F.	13.3	3.9

STRESS FROM DOUBLING OF
RATED WINDSPEED

FIGURE 5



	A	B
PRINCIPAL STRESS P.S.I.	0	10,585
MAXIMUM SHEAR P.S.I.	1913	0
MIN. S. F.	12.0	3.6

POLE MATCHER
ERECTION STRESSES

FIGURE 6

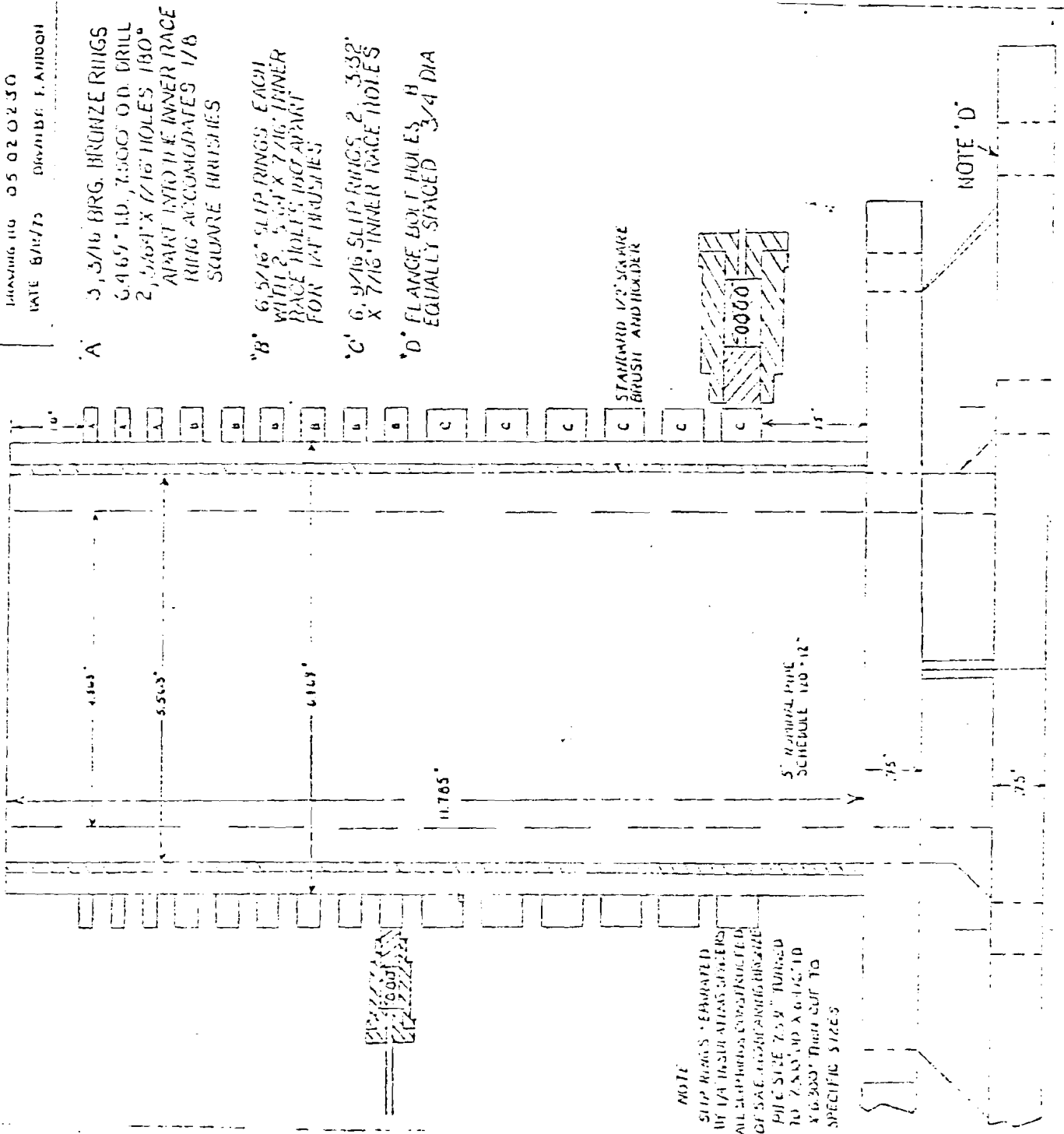
decisions made. It was decided that six 9/16" wide sliprings would be rated for high power capacity, six 5/16" wide for medium duty, and three 3/16" rings for instrumentation. Using $\frac{1}{4}$ " spacing between the rings, the completed stack is 9 5/16" high and is positioned 1 $\frac{1}{2}$ " above the lower polematcher bearing and one inch below the main bearing. The sliprings are made of SAE 660 bearing bronze with a 6 $\frac{1}{2}$ " ID and a 7 $\frac{1}{2}$ " OD. The 6.5" ID gave sufficient clearance between it and the polematcher standpipe for the embedment of transmission wires in fiberglass and resin fillers. Figures 7 and 8 show detail of the slipring stack. Each slipring has two wires attached to it for redundancy.

2. Construction

As mentioned earlier, before the construction of a part began, guidance toward efficient machining was sought from the machine shop foreman and incorporated into the final design. With a little experience on the part of the designer, the number of these discussions were reduced and time saved. One important consideration was that, if possible, the final machining on all parts would take place only after all the welding and gas cutting was performed. At times, this posed problems, but the end result was a better quality on individual part tolerances. If welding or gas cutting had taken place after the critical machining, the thermal stress as a result of the high temperature would have caused warping in the finished product. This problem surfaced in the construction of the

PROJECT: ...
 SHEET: ...
POLE WITCHER ASSEMBLY
 DRAWING NO. 05 02 0230
 DATE: 8/27/73
 DRAWN BY: F. AIRDOH

- 'A' 3, 3/16 BRG. BRONZE RINGS
6.469" I.D., 7.500" O.D. DRILL
2, 5/64" X 7/16" HOLES 180°
APART INTO THE INNER RACE
RING ACCOMMODATES 1/8
SQUARE BRUSHES
- 'B' 6, 5/16" SLIP RINGS EACH
WITH 2, 5/32" X 7/16" THINER
RACE HOLES 180° APART
FOR 1/4" BRUSHES
- 'C' 6, 9/16" SLIP RINGS 2, 3/32"
X 7/16" INNER RACE HOLES
'D' FLANGE BOLT HOLES, 8
EQUALLY STAGED 3/4" DIA



NOTE
 SLIP RINGS SEPARATED
 BY 1/4" INSURANCE SPACERS
 ALL SEPARATORS CONDUCTED BY
 OF SAE 1020 CARBON BRONZE
 FILE SIZE 7.5" THICK
 TO 7.500" ID X 0.125" ID
 X 6.300" THICK CUT TO
 SPECIFIC SIZES

SLIP RING DETAIL
 FIGURE 7

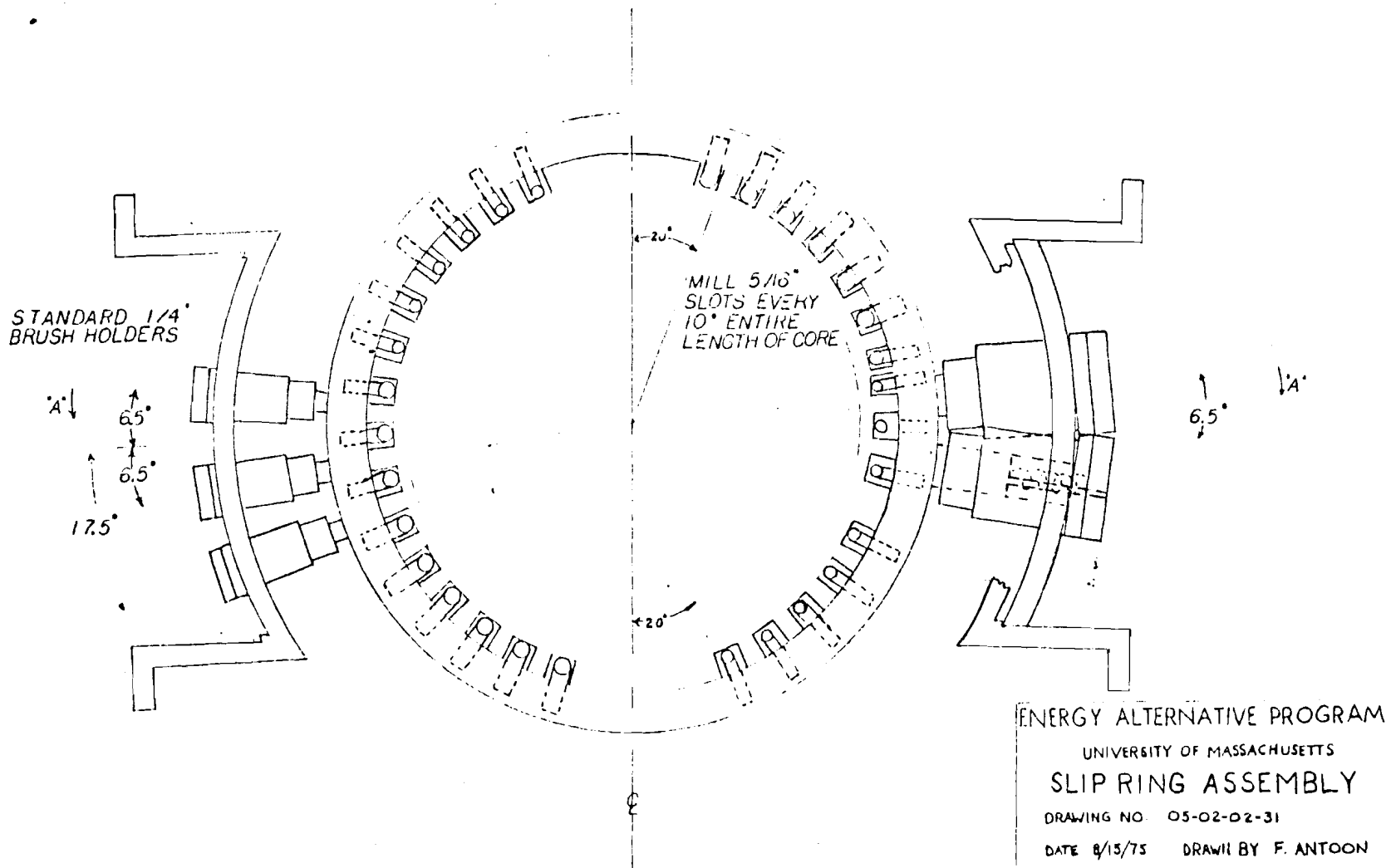


FIGURE 8

polematcher when in one afternoon, the entire assembly was welded together. The structure became noticeably warped even beyond the help of machining. It was found though, that by making use of a 60-ton press, the weldment could be restored to the designed geometry. Two machinists centers were placed on the top and bottom of the polematcher. It was through this axis that all final machining was accomplished. Now the polematcher is ready for the attachment of sliprings.

The standpipe OD is $5 \frac{9}{16}$ ", the slipring ID is $6 \frac{1}{2}$ ", in order to fit the sliprings the surface of the polematcher must be increased to a thickness of $\frac{15}{32}$ " using an insulating material, fiberglass matting, and epoxy resin. The insulator was built to just over $\frac{1}{2}$ " thick and then machined down so that the sliprings would have a snug fit and not slide easily. To the sliprings were soldered copper wires 180° apart. The heavy sliprings get #8 wire, the medium #14, and the instrumentation rings shielded coaxial cable. After preparing the slipring stack as in Figure 8, the sliprings were tapped and cemented into place with the electric power lines being threaded underneath the previously installed sliprings inside the $\frac{5}{16}$ " square channels. Once that cement had set, the polematcher was slowly turned on a lathe while a mixture of epoxy resin and thickening agent was poured between the sliprings. At the top and bottom of the stack, cardboard forms were used to retain the resin. A sufficient amount of resin was added until the slipring spaces were filled flush to the slipring OD. After the resin cured, the final machining was done. At the same time, the main bearing

seat and lower pitch bearing support were machined to specifications. The plastic pitchbearing turned to its final dimension and finally a light turning of the sliprings was done for polishing.

In order to prevent weather from blowing underneath the lower pitch bearing and into the slipring stack, each ribbed compartment of the polematcher was filled with foam and then sealed with a wood glue. A lightly greased main bearing added and the polematcher was complete. The machinists centers may or may not be removed at this point.

3. Cost

The following is the bill of polematcher materials and costs. The prices reflect late 1975 quotes.

BILL OF MATERIAL — WIND FURNACE

REV. 8/10/77

WORKING GROUP POLE MATCHER

SHEET NO. 1 of 2

DRWNGS: 05020201,05020203,05020205,05020220

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED COST	COST 1000
05020201 .01	POLE MATCHER STEM, 5" Nom. Sched. 120 5 9/16" O.D. with 1/2" wall	A. B. MURRAY SHARON, MA.	19"	\$32.00	
.02	POLE MATCHER BASE PLATE, (18" Dia. x 1" Thick)	DUANES' STEELYARD QUINCY, MA.	1	\$7.50	
05020203 .01	PITCH BEARING RING, (10" I.D., 12.2" O.D., 0.75" thick)	SHOP	1	\$22.00	
.02	POLE MATCHER RIBS	"	8	\$8.00	
.03	5.25" BORE TAPER ROLLER BEARING EQUIV. to a Tyson Cone #48290 & cup #48220	EASTERN BEARING FITCHBURG, MA.	1	\$63.58	25
05020205 .01	SLIP RING MATERIAL, (SAE660 brg. bronze)	MILLARD CO. BRAINTREE, MA.	13"	TOTAL COST \$115.00	
.02	" RINGS (7.5x6.5x9/16)	"	6		
.03	" " (7.5 x 6.5 x 5/16)	"	"		
.04	" " (7.5 x 6.5 x 3/16)	"	3		
05020220 .01	MAIN BEARING SUPPORT, (5.5" Dia. Nom.)	SHOP	1	\$5.00	
05020230 .01	HEX HEAD BRASS CAP SCREWS, (5/16-18 x 1.25)	ACCURATE FASTENER SO. BOSTON, MA.	36	@ 0.263 ea. \$9.47	
.02	FINISHED HEX BRASS NUTS	"	72	@ 0.078 ea. \$5.62	
.03	FLAT BRASS WASHERS	"	36	@ 0.058 ea. \$2.09	
.04	PLASTIC PITCH BEARING (PV-80) 13.5" O.D., 12" I.D., x 3/4" thick	T. F. E. PROVIDENCE, R. I.	1	\$25.00	

B. Main Frame

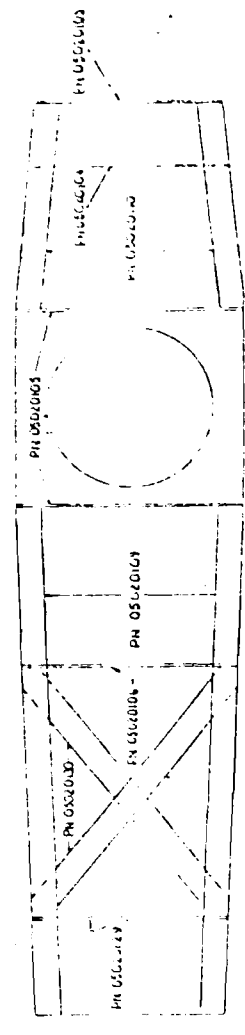
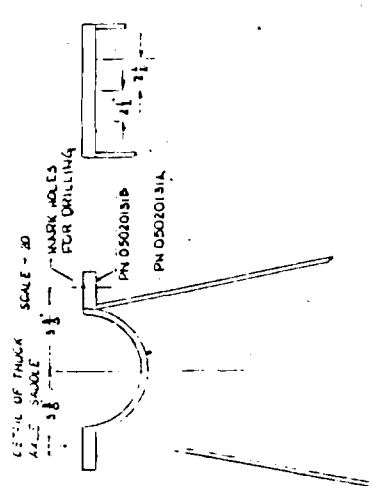
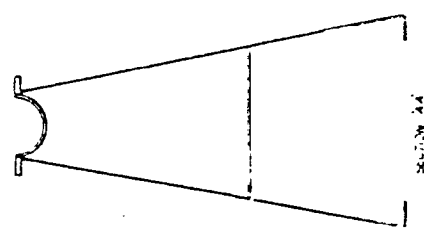
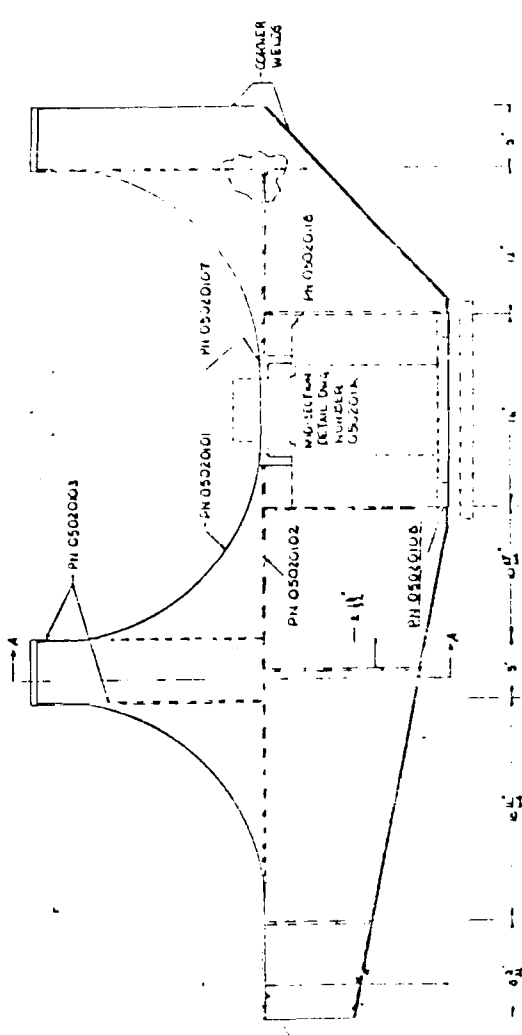
1. Design

The initial concept of the main frame is based on the general arrangement of the truck rear axle assembly and the generator. The basic parameters for the main frame layout is that it would support the truck rear axle assembly at the same positions a truck itself would mount. The assembly would be supported high enough above the top of the pole-matcher so that when converted to the WF-4, sufficient clearance would be allowed to make the ground shaft connection. The position of the generator should be far enough away from the hub and blades so that the machine would balance at the polematcher. For the first guess it was thought that 200# would have to be added at the location of the generator to balance the frame on the polematcher. This 200# was left as leeway against the fact that more accessories would be added there to help balance the frame. After the final weight and moment sheet was produced (Appendix), it was found that a net 342 inch-pound moment existed at the polematcher and that only nine pounds would have been required at the generator station to offset this. The balancing of this moment was due mainly to the addition of the yaw damper, lightning rod, oil and lube case, and the main frame center of gravity. This moment was considered so small that ballast was not added. As a result of the silent chain drive, a side moment of 3563 inch-pounds did exist. This is expected to be reduced somewhat when batteries and logic circuits are added to the opposite side of the main frame.

The type of construction for the main frame made very few changes to finally get to its present design, Figure 9. The first type considered was that using truss members formed out of angle iron. The design would have been light, but it was feared that not knowing exactly how the various accessories would be mounted, a design using flat plate was adopted. The large surface area of the plate steel offered many convenient locations for adding the accessories. Also, if a part of the main frame needed to be removed, it could be done so without jeopardizing the structural integrity. With a gas cutting torch, the frame sections would be easy to cut out as patterns from the main drawing. Some steel companies have a machine which will cut out full scale parts from the plan view of their respective drawings. Quotes were again gathered for the steel and are reflected in the cost tables.

Because the controlling dimensions from the polematcher and truck axle decrease width and height, a tapered appearance was chosen. Holes were cut in the sides of the frame to reduce weight and allow access to any component installed within the frame. Bulkheads were designed at certain locations to insure stiffness.

When the beam was designed, it was assumed that its strength was adequate. When the beam analysis was done only five critical sections were evaluated. Figure 10 shows those sections evaluated with Section "DD" being the most critical. Included in the analysis is the structural advantage of the truck assembly.



ENGINE ALTERNATIVE PROGRAM	
DATE	05/02/11
BY	
CHECKED BY	
APPROVED BY	

ALL DIMENSIONS IN INCHES
 TOLERANCES - 3/16
 SCALE - 40

MAINFRAME ASSEMBLY
 FIGURE 9

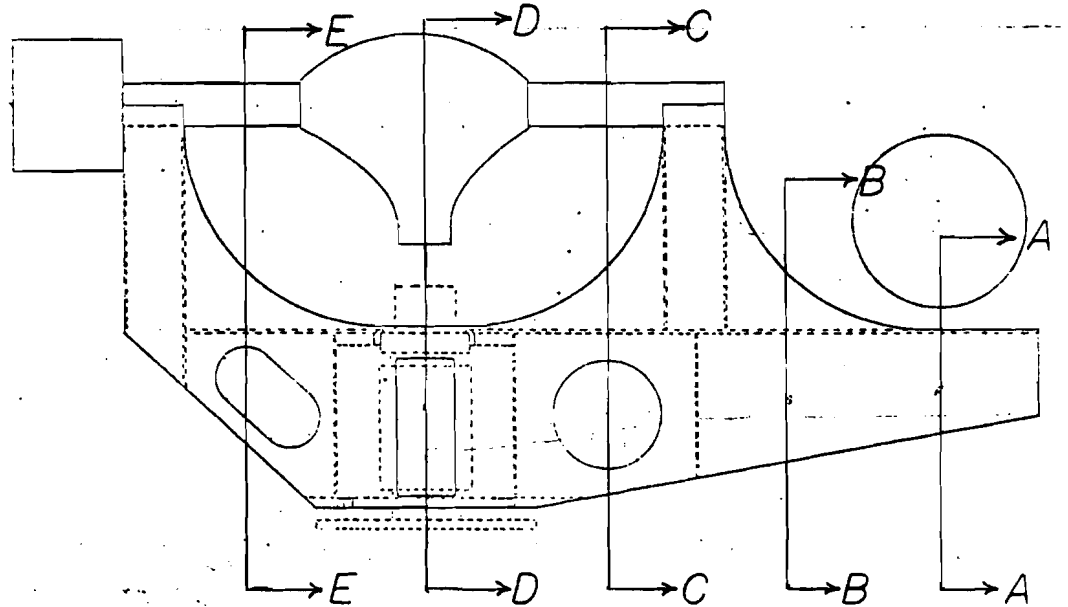
At section "DD" where the greatest bending moment occurs, a stress calculation was also done assuming the truck differential out of place for the normal loading condition, plus the machine erection condition, where the main frame wind axis is parallel to the ground (frame on its side). Figure 10 shows the results of the analysis for both 1/8 and 3/16" steel plate. The moments used to determine the section stresses are those taken from the weight loading in Figure 2 for the rated thrust loads.

Stress sections of the beam were not evaluated because of the extremely large area moment of inertia along that axis.

The final analysis concerns the force flow path into the polematcher. Of greatest concern is the area about the main bearing where the weight of the machine is carried and the greatest radial force occurs. Essentially the ribs take the entire load here. Figure 11 shows the worst case load and the corresponding compressive stress leaving a safety factor of about 2. For the lower pitch bearing, if one assumes this to be a bar point loaded as in Figure 12, the bending stresses leave a safety factor of about one. By looking at the structure around the assumed bar, one realizes that it does indeed contribute to the overall structural integrity and, therefore, must conclude that the safety factor is reasonably greater than one.

2. Construction

Except for one area, the construction of the main frame progressed well. The one construction difficulty was the concentric machining of the main and skirt bearing surfaces. This was done by setting the finished main frame on a horizontal



SECTION STRESS RESULTS

SECTION	AA	BB	CC	DD	EE
MOMENT DUE TO WEIGHT (N-LB)	904	6985	15015	29581	18882
^{3/16} " AREA INERTIA (IN ⁴)	428	839	1467	2012	1147
PRINCIPAL STRESS (PSI)	34	101	165	297	237
LESS DIFFERENTIAL (PSI)				1777	
ON SIDE (PSI) WO/DIFF.				566	
1/8" PLATE AREA INERTIA (PSI)	291	568	1178	1968	1139

FIGURE 10

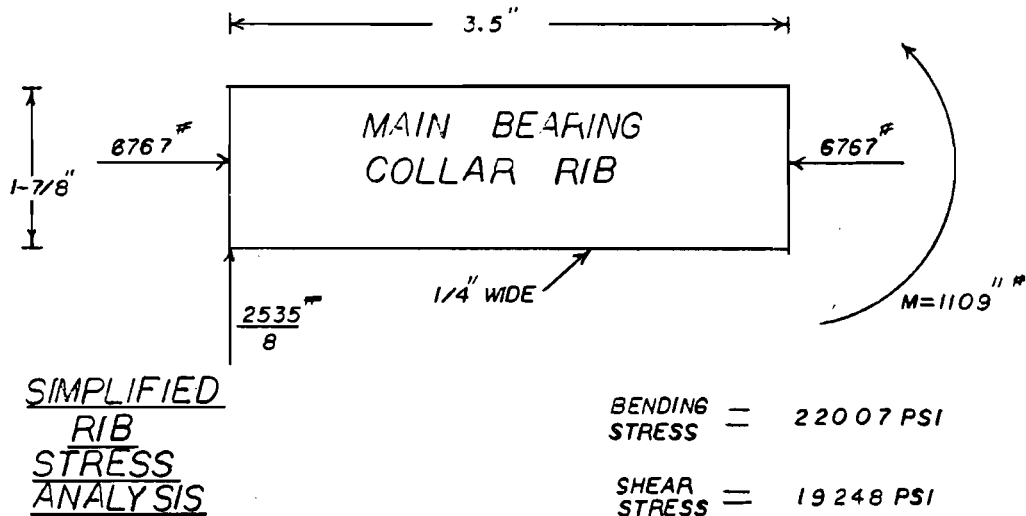


FIGURE 11

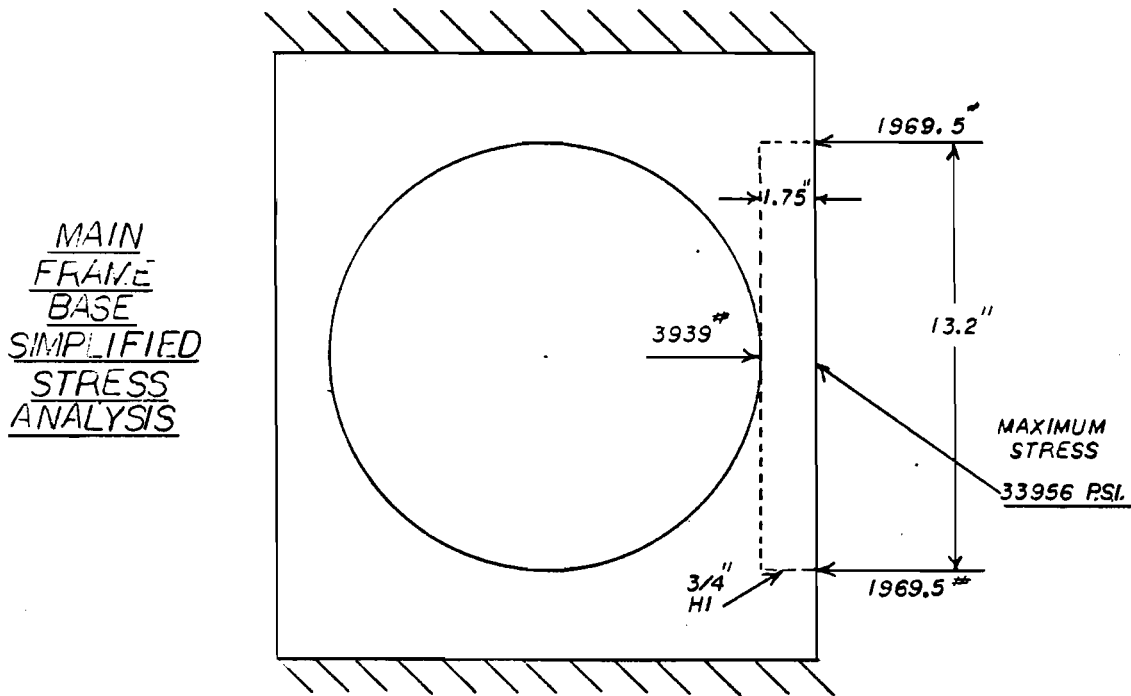


FIGURE 12

milling machine. The milling spindle was replaced by a crank and tool bit. Care was taken in the machining so that deflections of the crank could be held to a minimum in order that the tolerances could be held.

The frame was welded together in the following manner. After welding the ribs to the main bearing collar, the collar ID was turned to .1" under the finished dimension. The frame sides and bulkheads were cut out using both a cutting torch and table saw. The bulkheads were first attached to one side of the frame and then the other side was welded into place. The truck support saddle, main bearing collar, and frame base were added. Finally, the bearing surfaces were bored concentrically and then the frame painted. The polematcher and main frame were joined at this time.

The polematcher is bolted onto a large sturdy pallet. All of the polematcher-to-tower bolts should be used because once the frame is lowered onto the polematcher, there is not enough clearance to install them. The main bearing is lightly greased, installed and then the frame is lowered onto the polematcher. The main bearing has a very steep roller angle. It would take an extremely large side load to allow the bearing cup to slide up the face of the cone roller, therefore, the weight of the machine and its centroid being very close to the polematcher is the only hold down force required. During erection, it is quite possible for the machine to slip off the polematcher. To correct for this, two locking caps are screwed onto the polematcher, 1/32" above the main bearing collar, and then tightened against

each other for locking purposes.

Inside the main frame slipring housing, two brush blocks are bolted to the main frame base plate and 3/4" x 3/4" carbon brushes wipe over the top of the polematcher ring, which is Item 3 in Figure 3. This is for lightning protection, so that current does not flow through the infinitesimal contact area of the main tapered roller bearing. If this were to occur, the bearing could possibly weld itself together.

3. Materials and Costs

The following is the bill of main frame materials and costs. The prices reflect late 1975 quotes.

BILL OF MATERIAL — WIND FURNACE

REV. 8/10/77

WORKING GROUP MAINFRAME COMPONENTS

SHEET NO. 1 of 3

DRWNGS; 05020101,05020102,05020112,05020113,05020114,05020116,05020118,05020121

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED COST	COST 1000
05020101 .01	GENERATOR BEARING FOUNDATION PEDESTAL #3	MACHINE SHOP	2/8# stl	\$4.00	
.02	" " " FLANGES	" "	2/1# stl	\$1.00	
05020102 .03	LOWER PITCH BEARING OUTER RACE (16x18x1")	" "	1/55#	\$26.00	
05020112 .01	MAINFRAME DECK, 3/16" Plate Steel	" "	1/43#	\$22.00	
.02	FRONT LOWER BRACES, (2"x43"x3/16")	" "	2/10#	\$5.00	35
.03	REAR LOWER BRACES, (2"x22"x3/16")	" "	2/4.5#	\$2.25	
05020113 .01	MAINFRAME SIDES, 3/16" Steel Plate	" "	2/122#	\$61.00	
05020114 .01	MAIN AXLE SUPPORTS,(Top 12",Bot.11.9",Ht18.1)	" "	3/25#	\$12.50	
.02	MAINFRAME BULKHEADS,(" 12", " 17.4", " 11.8")	" "	2/22#	\$11.00	
05020116 .01	MAIN AXLE SUPPORT,(t5.1",b 13.3",h 22.3")	" "	1/11#	\$5.50	
.02	FORWARD BULKHEADS,(t 12",b 16.6",h 11.8")	" "	1/9#	\$4.50	
05020118 .01	MAIN BEARING BRACE, (9" O.D.,6.5"Nm,I.D)	" "	1	\$39.10	
.02	RIBS FOR MAIN BEARING SUPPORT	" "	6/3#	\$2.00	
05020121 .01	PINION BEARING BRACKET	" "	1/5#	\$3.00	

BILL OF MATERIAL — WIND FURNACE

REV. 8/10/77

WORKING GROUP MAINFRAME COMPONENTS
 DRWNGS: 05020121, 05020122, 05020124, 05020131

SHEET NO. 2 of 3

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED COST	COST 1000
05020121 .01	PINION BEARING SPACER	MACHINE SHOP	1/1#	\$1.00	
05020122 .01	" " BRACKET BASE	" "	1/3#	\$1.00	
.02	" " FOUNDATION STRAPS	" "	2/2#	\$2.00	
.03	" " " CROSS BRACE	" "	1/5#	\$2.50	
.04	" " BRACKET SUPPORT	" "	1/4#	\$2.00	36
.05	" " " "	" "	1/1#	\$1.00	
.06	" " FOUNDATION "	" "	"	\$1.00	
.07	" " " PEDESTAL	" "	1/4#	\$2.00	
05020124 .01	" " " FLANGES	" "	2/2#	\$1.00	
.02	" " " SIDES	" "	"	\$1.00	
.03	OUTBOARD GENERATOR BEARING BRACKET	" "	1/1#	\$1.00	
.04	INBOARD " " "	" "	"	\$1.00	
05020131 .01	BOLTS AND " "	" "	1/8#	\$4.00	
.02	GENERATOR BEARING BRACKET BASE	" "	1/2#	\$1.00	

BILL OF MATERIAL — WIND FURNACE

REV. 8/10/77

WORKING GROUP MAINFRAME COMPONENTS

SHEET NO. 3 of 3

DRWNGS: 05020131, 05020133, 05020134

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED COST	COST 1000
05020131 .03	OUTER GENERATOR BEARING BRACKET SUPPORT	MACHINE SHOP	1/2# stl.	\$1.00	
.04	" " " " "	" "	1/3# stl.	\$1.00	
.05	GENERATOR BEARING BRACKET SUPPORT #1	" "	1/4# stl.	\$2.00	
.06	" " " " #2	" "	"	\$2.00	
05020133 .01	INBOARD GENERATOR BEARING SPACER PLATE	" "	1/2# stl.	\$1.00	
.02	GENERATOR BRG. BRACKET FOUNDATION SIDES	" "	2/4# stl.	\$2.00	37
05020134 .01	" " " " SUPPORT	" "	1/2# stl.	\$1.00	
.02	" " FOUNDATION "	" "	"	\$1.00	
.03	" " " CROSS BRACE	" "	1/5# stl.	\$2.50	
.04	" " " PEDESTAL #1	" "	1/4# stl.	\$2.00	
.05	" " " " #2	" "	"	\$2.00	

C. Power Transmission

The idea of being able to vary the form of the energy transferred from the wind turbine to the ground was kept in mind throughout the design. The primary component that was considered to make the change of generating electricity aloft or sending shaft work to the ground was the truck rear axle assembly. Two questions had to be answered though before work with this concept could begin. The first question of cost was analyzed and found that the pricing of a speed drive of equivalent capacity to that already available in the axle assembly exceeded the cost of the axle assembly. Therefore, the immediate advantages of the axle assembly above the speed drive it housed were: the 90° shaft drive required for the WF-4, the supporting structure for the 90° speed drive assembly, the rotor spindle, rotor, high strength shafts, rotor bearings, and brake mechanism. The disadvantage of the axle assembly and second question to be answered was: could the assembly be modified to suit the needs of the wind turbine. Of major concern was drilling a 5/8" hole the entire length of the truck axle shaft for purposes of a blade pitching link and to be able to lock the differential gears so that the hypoid gear drive could be used. In order to help decide whether these modifications could be possible, a used comparable truck axle assembly was purchased from an automotive graveyard. Using this, it was determined that modifications were possible.

A new truck axle assembly was then purchased and used in the existing wind turbine, WF-1. Although the detailed design for WF-4 was not done, the total through process was, and is presented here.

1. Design

a. Wind Furnace-4 (WF-4)

The main frame, with the position of the truck axle pinion shaft above the center of the polematcher, has already been explained. WF-4 begins its change from WF-1 when the truck pinion shaft is rotated down 90° . When this is done, one finds that although the shaft axis is in line with the lateral axis off the center of the polematcher, the shaft axis is displaced about $1\frac{1}{2}$ " away from the longitudinal axis. This is the characteristic of the hypoid gear drive. One solution is to angle the shaft toward the hole in the polematcher, press the first bearing set into the top of the polematcher, bring a shaft up through that bearing with a splined end and attach the pinion shaft to that vertical shaft with one constant velocity universal joint. Another solution would be to drop the pinion shaft straight down and use a shorter shaft with two hooke type universal joints to connect the pinion to the ground shaft. With this method, the first shaft bearing may have to be placed lower into the polematcher in order to gain sufficient clearance for the universal joints and splined coupling. With either system a splined shaft interface would be required at the point where the tower and polematcher are fastened. In order to counter the yawing torque induced by the ground shaft, a yaw driver, possibly of a worn gear type should be used.

Running the ground shaft through the length of the tower would be done in stations consisting of one shaft, two universal

joints and one bulkhead bearing. The length of each station is determined by a calculation which determines the resonant frequency in shaft whirl. Shaft whirl is an unstable vibration condition which would impose high loads on the shaft, shaft bearings and universal joints causing them to fail prematurely. Using a nominal pipe of 1.66 OD and 1.20 ID, each shaft station must be no more than 7 feet apart. The calculation was done using an equation for shaft whirl, the above torque tube and the pinion rated rpm. If shafting were desired in a basement 68 feet from the wind axis, 68/7 or 10 bearing stations would be required.

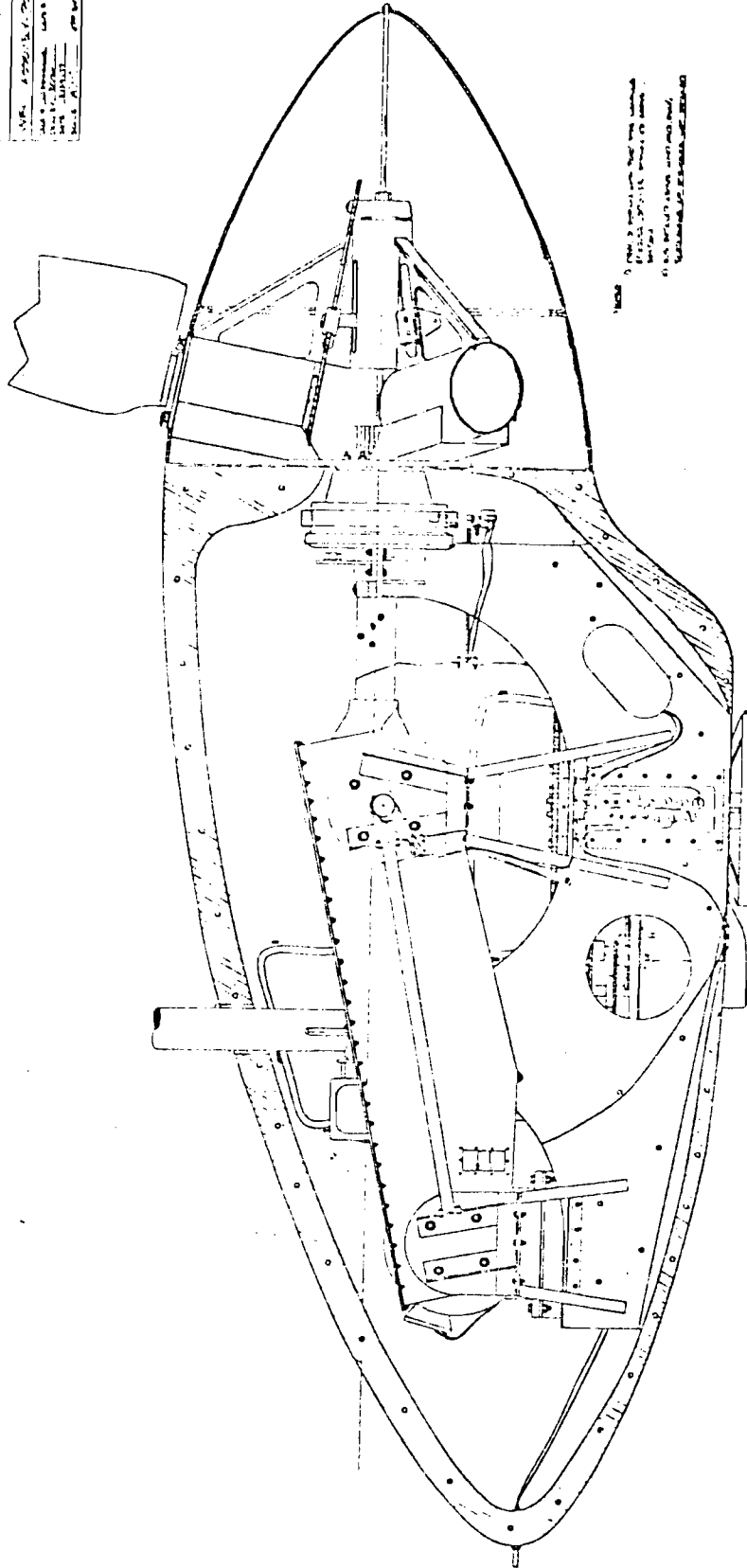
To install the shafting in the tower, one method would be to build the shafting, bearings, and universal joints outside the tower. The bearings would be fastened to bulkheads whose exact positions would be located on the tower. The bulkheads would have two threaded ends 180° apart, exact bolt holes would be located on the tower and then the bulkheads fastened inside. Alignment is not critical because of the flexibility offered by the universal joints. The existing WF-1 has a provision of bringing the shaft into a short tunnel beneath the tower's base. Here another 90° speed differential could be used to bring the shaft power away from the tower and into whatever power receptor is desired.

b. Wind Furnace-1 (WF-1)

The arrangement for WF-1 is shown in Figures 13 and 14. Here, shaft work from the wind rotor turns hypoid gears with a step up ratio of 4.86 to 1. That output shaft drives a

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DATE	10/15/50
BY	W. H. ...
REASON	...



Notes:
1. This drawing is for information only.
2. It is not to be used for construction.
3. It is not to be used for identification.
4. It is not to be used for reproduction.
5. It is not to be used for any other purpose.
6. It is not to be used for any other purpose.
7. It is not to be used for any other purpose.

FIGURE 13

set of silent chain sprockets with a step-up ratio of 2.22 to 1, which in turn drives the 25 Kw generator. For this design, the entire power unit is enclosed in the WF-1 nacelle and energy transferred to the ground by transmission lines. The design considerations discussed here were the modification of the truck rear axle and the selection of a second stage speed drive.

In order to be able to use the truck axle, three experiments would have to be successfully completed. The truck axle shaft is a solid, specially heat treated steel alloy with an OD of 1.35. The design for the WF-1 variable blade pitch requires a 5/8" diameter clearance through the center length of this axle. The problem is to drill this hole accurately for its 36" length. The next problem was to determine whether or not the differential gears could be successfully locked, providing us with a 90° hypoid gear box. The final task was to ensure structural integrity of the complete axle assembly.

To answer these questions a used, one ton, rear axle assembly was purchased, taken apart, and studied. A 5/8" hole was successfully drilled the entire length of the axle shaft even though the steel hardness did make the task more difficult, and the differential gears were successfully locked. After applying the estimated rated and weight loads of the wind turbine to this axle assembly, it was found that the spindle upon which the windrotor turns was structurally inadequate and the shaft torque carrying capability was questionable. A

slightly larger rear axle assembly was investigated and purchased new. Figure 15 shows the truck axle exploded view. An analysis was conducted on this rotor spindle using as loads the rotor weight and moment due to angular momentum of the spinning rotor. It was found that the highest stressed point on the spindle carries a safety factor of three. The differential gears were locked and this procedure appears in Appendix II. The design torque carrying capability of the hypoid ring gear was compared to the rated wind rotor torque of 1572 ft-lbs. A letter received from the gear manufacturer states a continuous rating of 1750 ft-lbs and a maximum rating of 7000 ft-lbs. A still questionable area of strength is the wind rotor shaft.

To determine its structural integrity, a non-destructive hardness test was conducted and information was gathered from the manufacturer concerning metal type and hardness.

The two widely used methods for determining steel hardness are the Brinell and Rockwell standards. It was learned that a hardness of Rockwell C48 existed at the surface and C20 from .15" to .24" below the surface. Although there is no direct relationship between hardness and yield or tensile strength of steel, a useful correlation does exist between the hardness of a material and the tensile strength, endurance limit, and wear resistance. The tensile strength of steel is usually about 500 times the Brinell Hardness number.* A Rockwell C48

* Ref. #7, p. 580.

REAR AXLE (SINGLE SPEED) -DANA (SPICER) 60, 60HD, 60.3E, 61-1 and 70B (NON-LOCKING)
1973 / E-F-P250/400

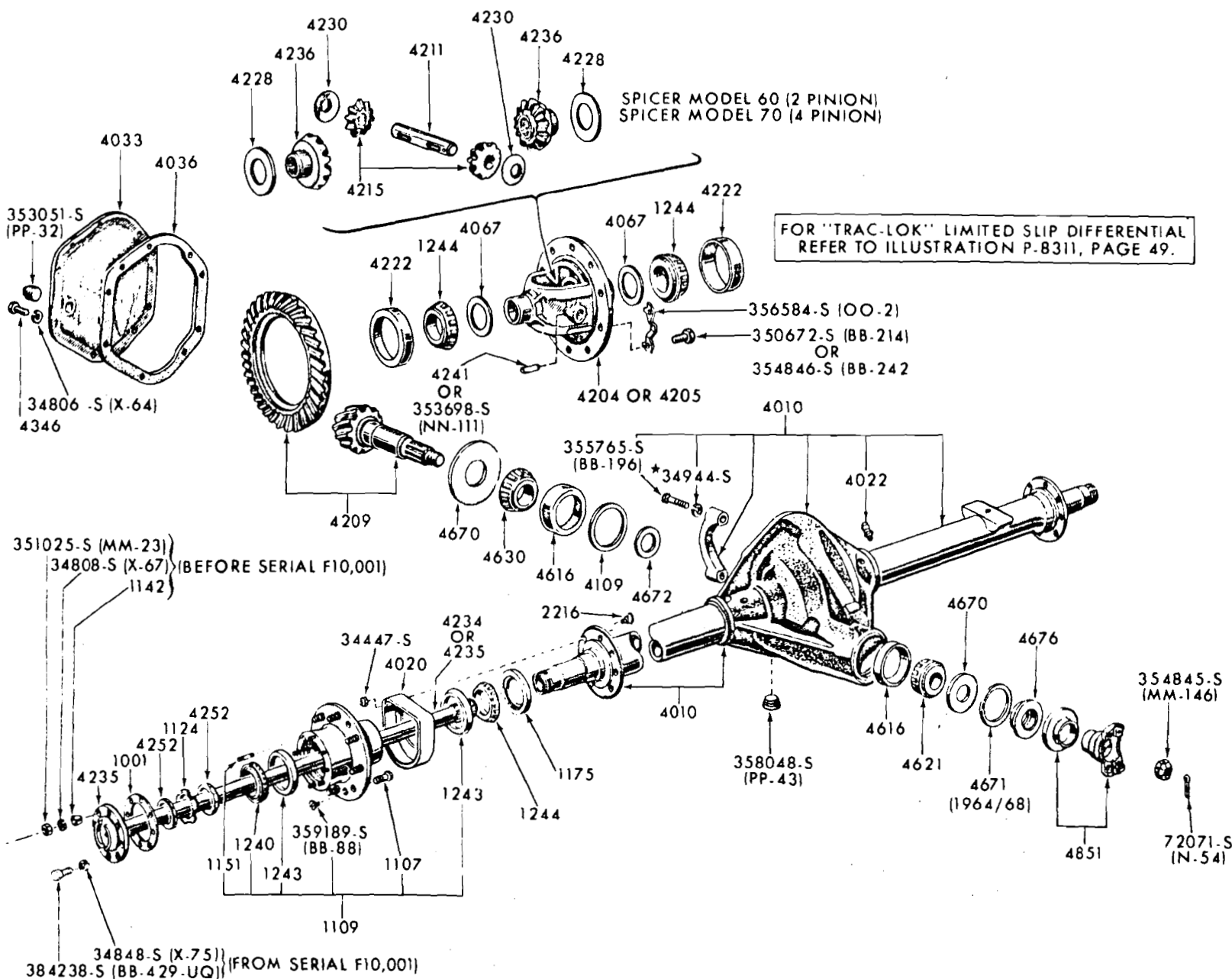


FIGURE 15

corresponds to a Brinell 460, using the approximation, the shaft surface can withstand about 230,000 psi of stress. This was verified by the axle supplier who states a working tensile strength of 232,000 psi. Because the material below the surface is not as hard, it would be expected that the shaft tensile strength is less. There is no correlation usually between tensile and yield strength of a material, but after inspecting the trends of other hardened materials and picking the worst cases the yield strength in shear of this shaft was estimated at 90,000 psi even though the supplier states the yield stress at 125,000 psi. The wind rotor torque would put a stress of 45,000 psi on this shaft, therefore, an estimated safety factor of 2 is present. During the shaft's time in operation its torque has been increased to 1.5 times the rated value. When the machine was taken down for overhaul, the shaft was inspected and there was no evidence of permanent yield. Because the shaft would be run at high cyclic loads, it was decided to have the shaft shot peened. Shot peening a steel shaft of this hardness has an effect of increasing the fatigue endurance limit by a factor of two. A reference substantiating this appears in the Appendix. In order to monitor shaft torque a set of strain gages were permanently attached to the windshaft along with sliprings. This was done so that shaft torque could be monitored directly and over stressing prevented.

Although the truck axle is being modified for the wind turbine, trouble was taken to mount and operate the assembly

in the WF-1 in the same normal manner as it would on an actual truck. So the axle does sit in an upright manner and the hub rotation corresponds to a forward vehicle motion. The principle of the drive corresponds to the vehicle rolling downhill using the engine and a lower gear for deceleration, so energy is flowing into the drive instead of out. With these restrictions and the fact that the blades rotate clockwise looking downwind, then the truck wheel hub to which the machine hub and blades were fastened was the left or driver's side of the vehicle.

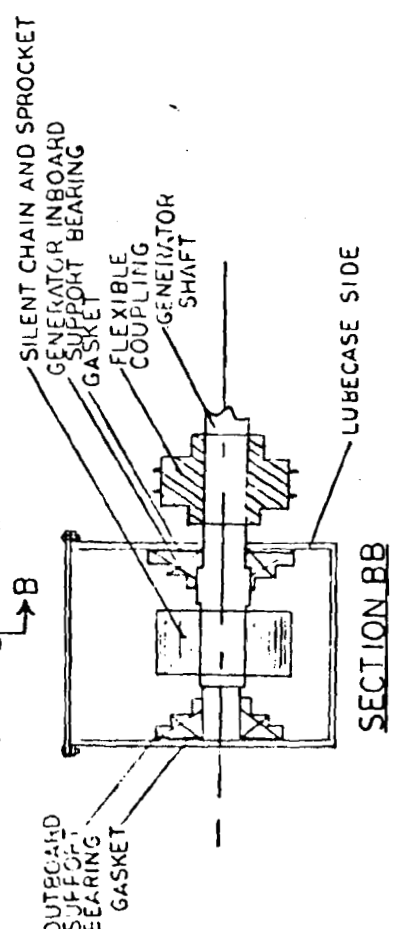
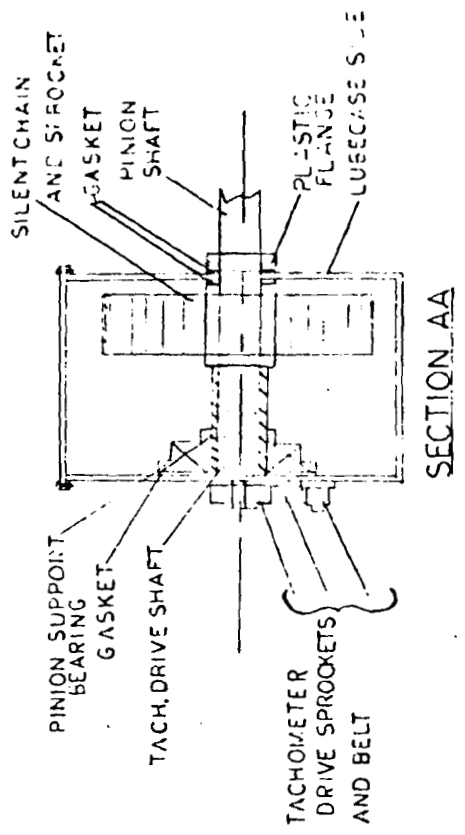
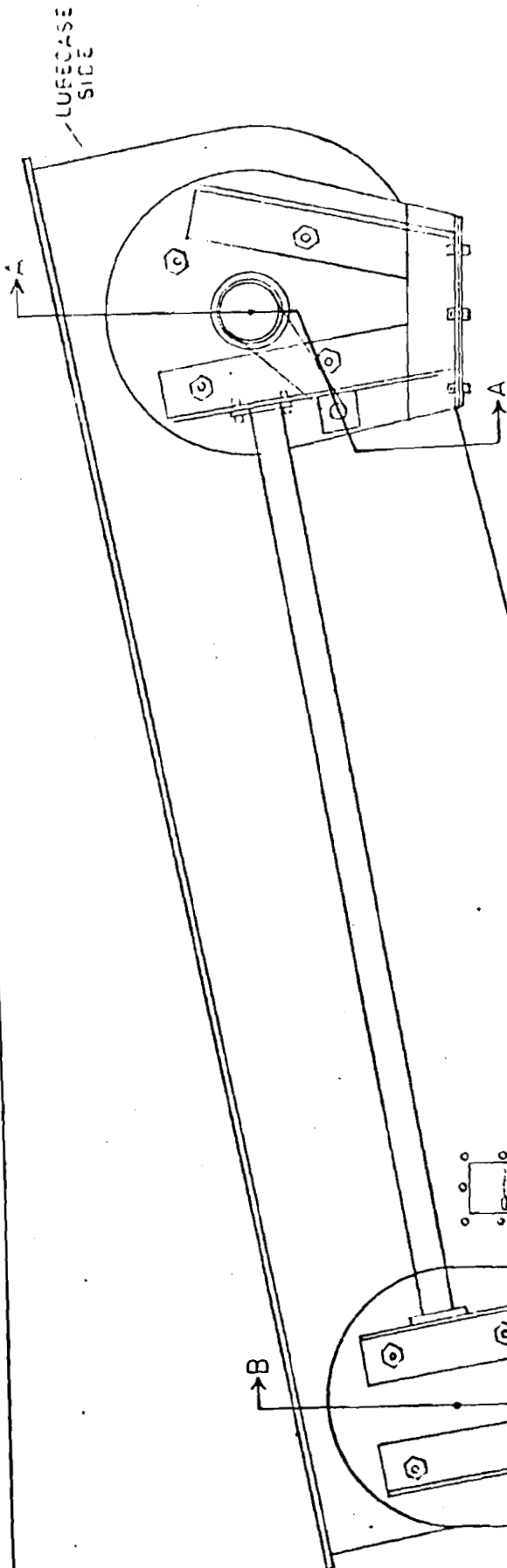
Continuing on with the power transmission design, a second stage speed drive was required. A main concern here was for that of an efficient drive, for that reason a V-belt drive was not chosen. Timing belt was considered but could not be used because a drive of sufficient power capacity was not available. The choices were narrowed to a gear box, roller chain, or silent chain drive. Silent chain costs more than roller chain, but when fitted to the WF-1 application, roller chain would experience a region of galling which causes the chain to wear at an increased rate. The unique design of silent chain inhibits galling, making it a better choice. A gear box drive was selected, but found to have an expense three times that of silent chain; hence silent chain became the accepted choice. Along with the acceptance of the silent chain was the requirement for a lubrication system. The main considerations here were that of sealing the case from oil leaks, and to keep to a minimum the lateral width of the machine.

Figure 16 shows the detail of lube case assembly. The bearings used, all have their own oil seals so no others were required. Throughout the bench testing and up to three months after WF-1 was installed, the seal functioned correctly. After that time, one seal began to leak, but soon corrected itself.

2. Construction

The final assembly has been shown in Figures 13 and 14. The truck axle assembly was seated into fiberglass and epoxy resins at the main frame saddles. This was done to dampen vibrations being transferred from the rotor. Other modifications were made of the rear axle assembly for the purpose of variable blade pitch. These modifications were not complex and are described in another report, "Design of Hub and Pitching System for the Wind Furnace I", UMass, June 1977. The following paragraphs discuss those assemblies which are not obviously shown through the design drawings.

One major time consuming task was that of paralleling and truing the silent chain sprocket shafts with respect to each other. This was done using a recommended procedure supplied by the manufacturer of the sprockets and the guidance of the engineering department's machine shop foreman. The basic procedure entailed truing the shafts using dial indicators, leveling with a standard fluid level and paralleling by constructing right angles from the shaft axles and setting an exact distance of 41 3/4" between the shaft ends. The 41 3/4" is the recommended shaft centerline distance, which



ENERGY ALTERNATES
 U MASS
 PINION AND GENERATOR
 SHAFT ASSEMBLY INSIDE LUBECASE
 DRWG NO 0502012

FIGURE 16

was close enough to the designed 42". Silent chain, by design, allows flexibility in centerline distance of up to an inch, therefore, the shaft centerlines are not adjustable. Another major task was that of building and fitting the pinion shaft, Figure 17. Although the shaft could have been built in one piece, two separate sections, the splined assembly and the shaft body were built and welded together. The point at which the two were welded is on the 3 $\frac{1}{2}$ " shaft body, $\frac{1}{2}$ " away from the spline assembly. The spline dimensions were measured directly from the male differential splines. A problem was encountered when fitting the pinion sprocket shaft to the differential pinion. Apparently the bearing surfaces within the differential were slightly cocked from a vertical plane parallel with the longitudinal axis, causing a shaft runout of $\pm .01$ ". To remedy this, index marks were used to match the same splines and the seat of the splines was hand filed to a point where the introduced error offset the differential bearing error. The shaft runout was eventually reduced to $\pm .002$ ". The following is the procedure used to complete the construction of WF-1.

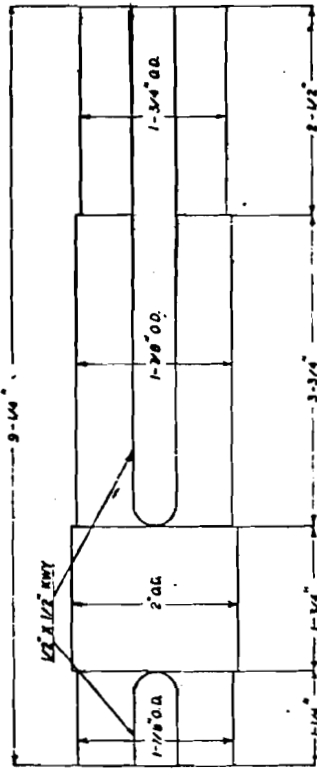
All bearings used were self-aligning and modified by tack welded mounting nuts to the installation holes. This was done so that if the nut were to loosen, it could not fall into the lube case and so that the bearing fasteners could be tightened without having to remove the lube case cover. The entire drive was assembled using large C-clamps as fasteners without the lube case so that the shafts could be positioned and trued. The exact bolt holes were drilled and those holes transferred to the lube case. In the final assembly the pinion sprocket shaft was bolted to the differential pinion.

NOTES: A MACHINE SHAFT O.D.'S FOR SLIDING
FIT WITH SUPPLIED PARTS UNLESS
OTHERWISE SPECIFIED

B UNLESS SPECIFIED TOLERANCES $\pm 1/32$ "

C SCALE FULL

GENERATOR SPROCKET SHAFT
P N 05020315



MAKE 1/2
SQUARE KEY
FOR ABOVE
SHAFT



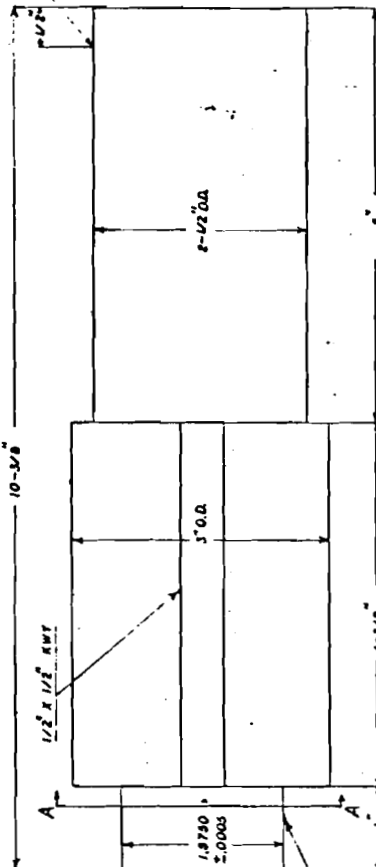
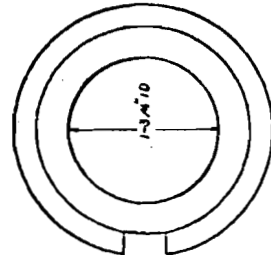
1/4 RADIUS
BRIND TO FIT
ABOVE KEYWAY



Broach splines
to fit
differential
pinion shaft

POLISH SURFACE
FOR OIL SEAL

DRILL AND TAP FOR
5/16 X 1/8 SETSCREW



PINION SPROCKET SHAFT
P N 05020316

ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
GENERATOR + PINION SPROCKET SHAFTS	
DATE	14 SEPTEMBER 1976
DRAWN BY	A. J. J. ...
05020320	

FIGURE 17

Next, the gaskets, sprocket, bearing, lube case, and bearing support were slid into that shaft at one time. All bolts and set screws were put into place, but not fastened down completely. At the generator end, the bearing supports, gaskets and bearings were loosely fastened into place, and then the generator extension shaft slid through the bearings and into the generator sprocket. A final shaft runout was checked and all fastenings were tightened. The generator was brought into place and shimmed to align its shaft with the generator extension shaft, then a roller chain coupling was added. Initially a rigid coupling was used, but due to the fact that some vibration was present, it got hot and so a decision was made to use the flexible chain coupling. The silent chain was installed and type "A" automatic transmission fluid added to a level where through the sight glass, contact was made with the chain's lower strand; about 9 quarts. Two and one half quarts of hypoid gear oil was added to the truck differential. The lube cover was added, a side strut installed, and the machine was ready for the bench tests.

3. Bills of Material and Cost.

The following is the bill of transmission materials and costs. The prices reflect late 1975 costs.

BILL OF MATERIAL — WIND FURNACE

REV. 8/10/77

WORKING GROUP GEARING & LUBECASE

SHEET NO. 1 of 2

DRWNGS : 05020321

ITEM	DESCRIPTION	SOURCE	QUANTITY	ESTIMATED I	COST 1000
05020321 .01	SILENT CHAIN LUBRICATING CASE	SHOP	1	\$30.00	
.02	SHAFT OIL SEAL, 2.5" Diameter		2	\$4.00	
.03	" " " , 2" "		2	\$4.00	
.04	RAMSEY SILENT CHAIN SPROCKET, 31 Teeth 1/2" Pitch x 3" wide, 1 7/8" Bore, 1/2 x 1/4 key	HOOKER TRANSMISSION CHICOPEE, MA	1	\$92.69	
.05	SILENT CHAIN SPROCKET, 1/2" Pitch x 3" wide 69 Teeth, 3" Bore, 1/2 x 1/4 keyway	"	"	\$177.19	54
.06	SILENT CHAIN CENTER GUIDE 1/2" Pitch x 3" wide	"	10'	\$187.00	
.07	CONNECTING LINKS FOR SILENT CHAIN	"	2 SETS	\$3.14	
.08	LUBE OIL FOR SILENT CHAIN, (Type A atf)		10 qts.	\$7.10	
.09	BOLTS FOR LUBECASE, 3/16"		46	\$16.00	
.10	PINION SUPPORT BEARING, (Type FB-350-2.5")	BROWNING CATALOG HOOKER TRANSMISSION	1	\$48.03	
.11	(Type FB-350-2") GENERATOR BEARING INBOARD SUPPORT	"	"	\$29.57	
.12	(Type FB-350-1.75) " " OUTBOARD "	"	"	\$25.35	
.13	PINION EXTENSION SHAFT	SHOP	"	\$25.00	
.14	GENERATOR " "	"	2	\$10.00	

VI. BENCH TESTING

For the purpose of the bench tests, the completed wind turbine was bolted to the floor of the test lab and connected to a 150 HP Ford diesel engine. The energy from the diesel was transmitted to the wind shaft by way of a torque pickup, extension shaft, and universal joints. The purpose of these tests were to allow a controlled break-in period for the new mechanical parts, to isolate and correct undesirable operational characteristics of the first and second stage drives, and finally, to determine the overall drive efficiency.

The tests first began in August. At this time the static torque required to begin the machine turning was 52 ft-lbs. After the completion of the tests and some 30 hours of operation, this torque value decreased to 27 ft-lbs. It is expected that under normal conditions this value will decrease slightly, but in colder weather, the thickening oil may increase it. After the initial runs, the rigid coupling connecting the second stage speed drive to the generator was replaced with a flexible one. This not only solved the heating problem, but reduced the drive vibration almost to nonexistence. Oil leaks were detected along the top of the oil case and around the fiberglass flange located at the differential housing. By the end of the test period, all oil leaks had been fixed except for the one at the differential fiberglass flange. The leak progressed at about 4 drops/hr. and was determined to be tolerable. This problem can be corrected, it was thought later, by the addition of a splash boot connected inside the lube case. It was mentioned earlier that although no other leaks developed on the test stand, after three months in operation, a leak did develop at the inboard generator sprocket bearing, but after more time in service, this problem corrected itself.

During the test, it was found that under full load, the oil temperature in the differential rose to 250° F. It should be noted that usually these differential assemblies are mounted to moving vehicles and that motion creates a cooling breeze. For the WF-1, two things were done to correct this. A positive ventilation system was installed and the conventional hypoid gear oil replaced with a synthetic type having many times the film strength. The ventilation system entailed installing radial fins to the windrotor in between the stationary and rotating nacelles. A vent hole was drilled from the inside of the main nacelle into that area and a ram air duct fastened near the tower on the bottom of the main nacelle facing upwind. In the lab, the synthetic oil reduced the temperature by 30°F over the conventional gear oil, and once installed the ventilation system will also yield a contribution. It should be considered that although the bench test was conducted at a full continuous load, the wind turbine will operate a majority of the time at a partial load. Therefore, the heat buildup rate will be less. All things considered, no problem is now expected in this area. The drive efficiency was determined by measuring input torque and speed and comparing that to generator output Kw for different speeds. From these comparisons, the drive efficiency including generator efficiency was found to be 82%. The bench tests were completed in October, 1976 with the mechanical drive accepted for full-rated operation.

VII FUTURE CONSIDERATIONS

As the building of the wind turbine progressed, modifications of the original designs were brought up and usually incorporated into the existing machine. Some specific modifications were either realized too late in the building stage or learned from the finished product, in which case, they could not be incorporated. Some specific considerations are discussed below.

A. Slipring Assembly

By reducing the size of the large sliprings; more, but smaller rings would be available. The brush blocks should be redesigned to fit inside the slipring housing possibly on insulated posts screwed into the main frame base. A new brush of rectangular shape could be used to achieve the same brush to slipring contact area. The advantage would be more sliprings, easier individual adjustment of brushes, and better protection of the brush holders which would be totally enclosed inside the housing. Much of the slipring expense came from the bronze rings having a $\frac{1}{2}$ " wall. Although the expense was less than that of a commercially available slipring stack, a better approach would have been to roll individual rings with a $\frac{1}{16}$ " wall into hoops, brazing the ends together. This would lead to a lighter, smaller and considerably less expensive assembly. There would be an estimated savings of 80%.

B. Main Frame.

Using $\frac{1}{8}$ " steel plate would be a savings in weight and cost with very little sacrifice in structural integrity. Instead of continuous welds, spaced 2" task welds could have made the fabrication less expensive. The present frame was built with a large surface area so that the fastening

of other machine accessories and supports would be made easier. Now that the location of specific components are identified, a simpler frame using steel channel or I-beams could save weight and expense. It is important to mention that much can be gained by forming a frame of fiberglass or wood. Once specific building techniques are identified, the individual pieces could be built without the need of expensive machines, materials, and manpower.

C. Transmission

One concern here is the bulky silent chain drive and the associated oil leakage precautions. The ideal drive would be the second 90° gear box, but now that is at least 3 times the expense of the silent chain drive. What could be considered though would be a second hypoid drive similar to that housed in the truck differential. It might be possible for the axle manufacturer to supply only the differential assembly at a reduced cost. Because these parts are mass produced anyway, it is very possible that this drive could cost less than the associated silent chain drive. It is possible that the redesign for such a drive would increase the height of the wind axis above the polematcher, but it would decrease the lateral width of the machine significantly. Probably the greatest headache though was the questioned structural integrity of the main wind shaft. Although a safety factor of two in yield and even greater in failure was assigned, it was not determined through direct engineering analysis. Much was dependent on the material hardness and/or assumptions. Because the shaft was being operated at such high stresses, a certain uneasiness was left with many of the project engineers. The shaft has shown its structural integrity in high instantaneous loads, but time is still required to determine the resistance to fatigue. It is recommended that safety factors of at least five be used in future designs.

VIII. CONCLUSION

It is now 19 months since WF-1 was put into operation. In the 18th month the machine was brought down for a general overhaul. The generator coupling had failed and has been replaced with a better one. Also, the sliding motion of the pitch linkage through the wind shaft acted like a pump, forcing the differential lube oil out into the nacelle spinner. This has been corrected with an oil seal arrangement in the nacelle spinner. It has been calculated that in an overspeed condition, wind shaft torque rose to 1.5 times the rated value and when that shaft was later inspected, there was no evidence of permanent yield along its 36" length. Because of the safety factor of 2, the lack of damage to the wind shaft seems to indicate no adverse torque transients throughout the power train dynamics. Other minor problems were present, but they did not hamper the operation. During the 18 months of operation, useful energy had been extracted from the machine and with the modifications to those problem areas mentioned above, it will continue to operate effectively. The WF-1 is making a contribution to the field of windpower, by the fact that it does its job well. It is by no means a perfect machine, but it is exemplifying many different designs and concepts, some of which being presented by this report. It is hoped that the reader can use this report as a basis in his own wind turbine design. He will make the decision on whether a certain concept is sound or not and then incorporate that toward what he feels is a better design. The ultimate goal is to make wind energy a commercial energy source; this project is a small step in that direction.

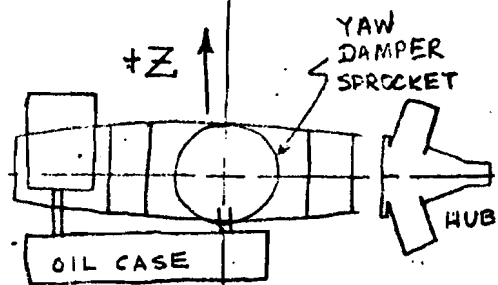
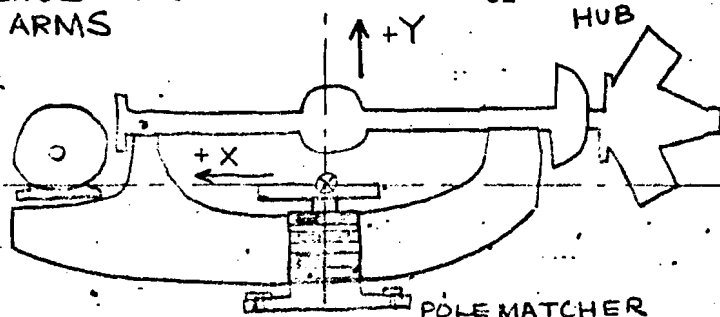
APPENDIX I

Weights and Moments

REFERENCE FOR
LEVER ARMS

62

GENERATOR



ITEM	WEIGHT LBS	VERT. LEVER INCH	VERT. MOMENT IN LBS	FORD LEVER INCH	FORD MOMENT IN LBS	STAB LEVER INCH	STRI MOMENT IN L
MAIN FRAME	322	-7	-2254	9	2898	0	0
POLE-MATCHER ASSEMBLY	153	-10	-1530	0	0	0	0
TRUCK REAR W/ BRAKE DRUM	232	16	5168	-6	-1938	-2	-64
FORWARD HOLD-DOWN CAP	6	17	102	21 1/2	129	0	0
AFT " " "	6	17	102	-22 1/2	-135	0	0
FORWARD LIFTING BRACKET	8	19	152	21 1/2	172	0	0
AFT " "	8	19	152	22 1/2	180	0	0
DIFFERENTIAL STRUT	2	7 1/2	15	7	14	-6 1/2	-13
BRAKE PARTS	5	16	80	-28	-140	0	0
PINION SHAFT :	6	16	96	0	0	-14 3/4	-88
" SPROCKET	47	16	752	0	0	-14 3/4	-69
" OUTBOARD BEARING	14	16	224	0	0	-18	-25
" " BRACKET	9	13	117	0	0	-19 3/4	-178
" BRACKET FOUNDATION	18 1/2	-4	-74	0	0	-14	-25
GENERATOR SHAFT :	4	8	32	41 3/4	167	-14 3/4	-59
" SPROCKET	13	8	104	41 3/4	543	-14 3/4	-192
" INBOARD BEARING	9	8	72	41 3/4	376	-11 1/4	-10
" OUTBOARD BEARING	5	8	40	41 3/4	209	-18 1/4	-91
" INBOARD BRACKET	9 1/2	6 1/2	62	41 3/4	397	-10	-95
" OUTBOARD BRACKET	9 1/2	6 1/2	62	41 3/4	397	-19 3/4	-187

ITEM	WEIGHT LBS	VERT. LEVER IN	VERT. MOMENT IN LBS	FORD LEVER IN	FORD MOMENT IN LBS	STBD LEVER IN	STBD MOMENT IN LBS
GENERATOR SHAFT BRACKET FOUNDATION	17	-3	-51	41 ³ / ₄	709	-13	-221
CHAIN	23	12	276	21	483	-14 ³ / ₄	-339
OIL CASE	65	12	780	21	1365	-14 ³ / ₄	-959
OUTBOARD BRACKET STRUT	8	12	96	21	168	-14 ³ / ₄	-158
FLEXIBLE COUPLING	4	8	32	41 ³ / ₄	167	-10	-40
GENERATOR	292	9 ¹ / ₄	2701	41 ³ / ₄	12191	3 ¹ / ₂	1022
" FOUNDATION	16	-2	-32	41 ³ / ₄	668	3 ¹ / ₂	56
BRUSH BLOCK ASSEMBLY	4	-11	-44	0	0	0	0
HUB AND BLADES ASSEMBLY	572	16	9152	-39 ¹ / ₂	-22594	0	0
PITCH CONTROL LINKAGE	32	16	512	20 ¹ / ₂	656	0	0
" " MOTOR AND GEARBOX	46	16	736	30	1320	4	184
" FEEDBACK LINKAGE							
BRAKE ASSEMBLY	35	5 ³ / ₄	201	9 ⁷ / ₈	346	-1	-35
" RE-SET PULLEY	2 ¹ / ₂	3/4	2	0	0	0	0
YAW-DAMPER ASSEMBLY	86	-5	-430	13	1118	2	172
" " SPROCKET	25	-1/2	-13	0	0	0	0
LIGHTNING ROD AND PIPE	80	60	4800	21 ¹ / ₂	1720	0	0
" " COLLAR	6	27	162	21 ¹ / ₂	129	0	0
TOTAL MAIN NACELLE	260	6	1560	0	0	-1/2	-130
SPINNING SHROUD	28	16	448	-36 ¹ / ₂	-1022	0	0

APPENDIX II

Locking the Truck Differential

Locking the Truck Differential

This procedure pertains only to the locking of this particular truck differential since there is a variety of different truck differential gear assemblies, and for each a unique locking method required. Reference to Figure 15 and its numbered parts are made.

Differential gears No. 4215 were set up to mesh with gears No. 4236 so that the tooth of gears 4215 was straight up and down within the valley of gears 4236. The adjacent valleys of gears 4236 were completely filled by consecutive passes of a welding rod, four valleys per gear. This restricted the gears from meshing at all. Next, pin 4211 was replaced with two studs, one end threaded into differential assembly 4204 and locked by set screws. The other end had a hole drilled and tapped; a bolt and washer large enough to retain gear 4215 was then installed. The center section of pin 4211 had to be removed to provide space for the blade pitching link which passes through the center of the differential.

REAR AXLE (SINGLE SPEED)-DANA (SPICER) 60, 60HD, 60.3E, 61-1 and 70B (NON-LOCKING)

1973/ E-F-P250/400

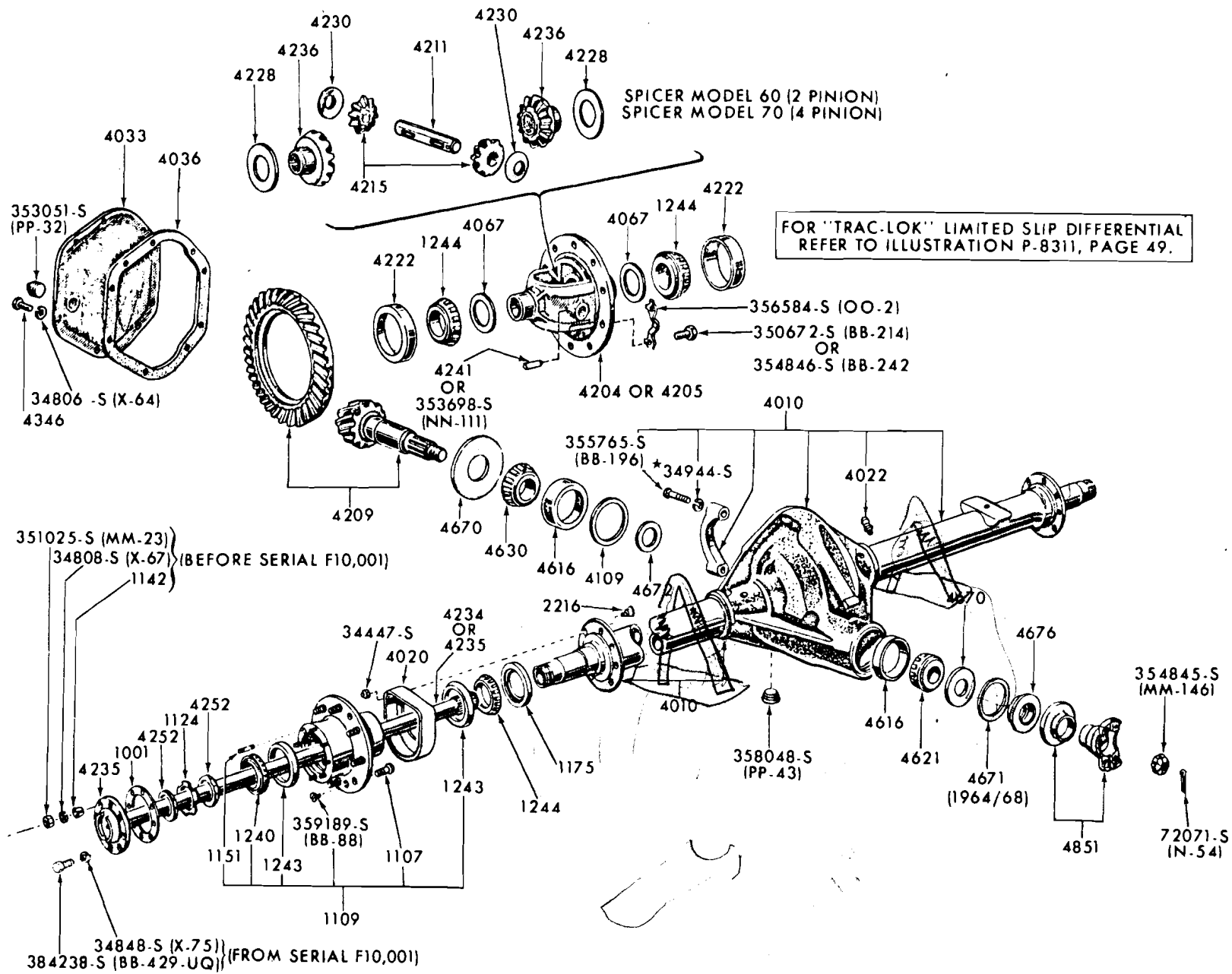


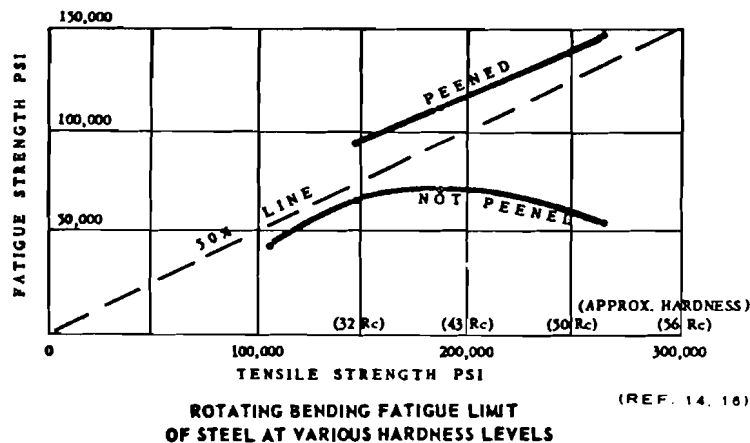
FIGURE 15

APPENDIX III

Shot Peening Very Hard Steel

SHOT - PEENING VERY HARD STEEL

In recent years increasing use has been made of steel at hardness levels above Rockwell C 50, corresponding to tensile strength above 250,000 psi, for fatigue and shock applications such as springs or aircraft landing gears. Such parts are always shot-peened in all critical areas. The chart below shows the close connection between shot-peening and the use of these strength levels:



Without shot-peening the greatest fatigue strength was obtained at a hardness around Rockwell C 40. At higher hardness the material lost fatigue strength (probably because of increased notch sensitivity). With shot-peening the fatigue strength increased continuously with greater hardness. At a hardness of Rockwell C 52 the shot-peened fatigue strength was 144,000 psi, more than twice the best unpeened fatigue strength.

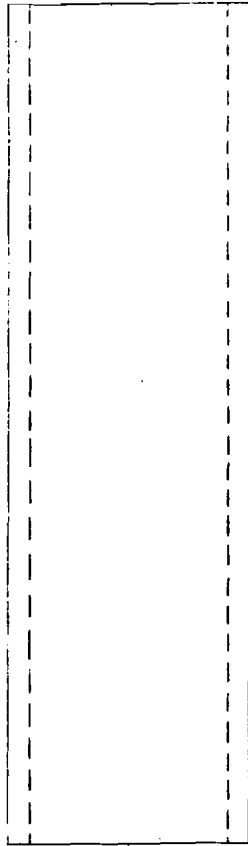
The chart shows that it is not sufficient to check the effect of peening at the conditions which were best without peening. Much larger improvement may be obtained by taking full advantage of peening with increased hardness.

The most interesting implication of these data is the increase of static strength made possible by the use of peening: Strength levels above 200,000 psi without peening are dangerous because of loss of fatigue strength. With peening the strength level can be raised both for static loads and for fatigue loads by using higher hardness. This practice has been used successfully in progressive industries.

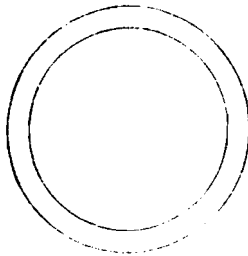
More recent tests (Ref. 14) have confirmed these data and added the knowledge that the fatigue strength of parts with peened surfaces is not impaired by shallow scratches which would ruin unpeened parts of high hardness (compare page 15).

APPENDIX IV

Remaining Project Drawings



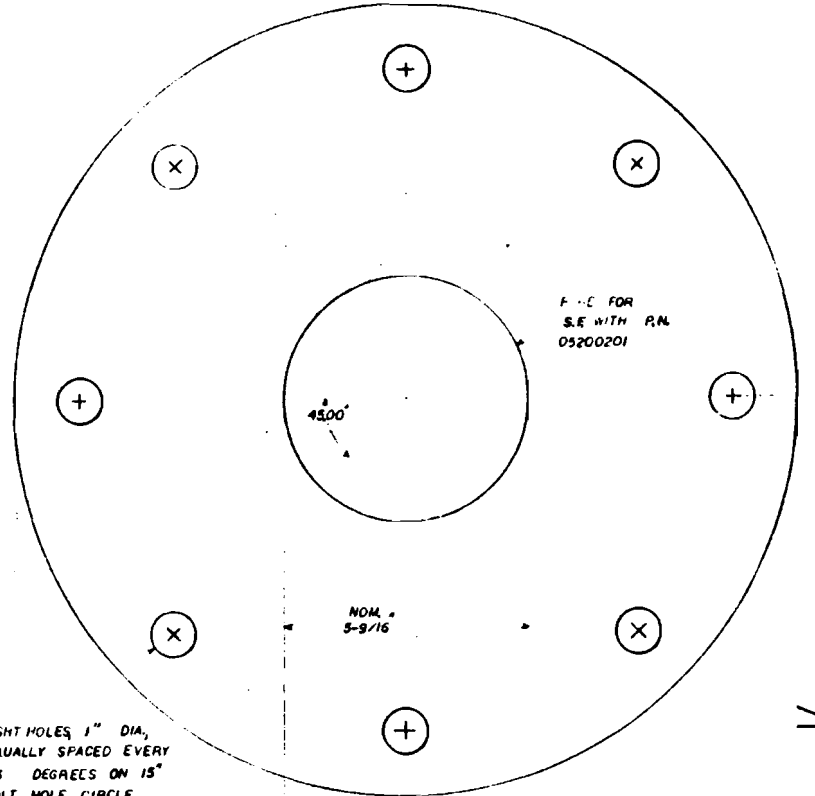
19"



ITEM 1 - POLE MATCHER STEM

PART NO. 05.02.02.01
 3 INCH SCHEDULE 120
 3.563 OD 4.563 ID .500 WALL
 GRADE A106 PIPE

ALL DIMENSIONS UNLESS
 OTHERWISE SPECIFIED
 ± 1/16" FRACTIONAL
 ± .01" DECIMAL



FACE FOR
 S.E. WITH P.N.
 05200201

45.00°

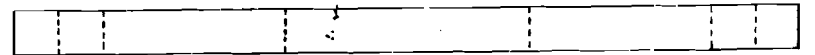
NOM.
 5-9/16

EIGHT HOLES 1" DIA.,
 EQUALLY SPACED EVERY
 45 DEGREES ON 16"
 BOLT HOLE CIRCLE.

ITEM-2 P.M. FACE PLATE

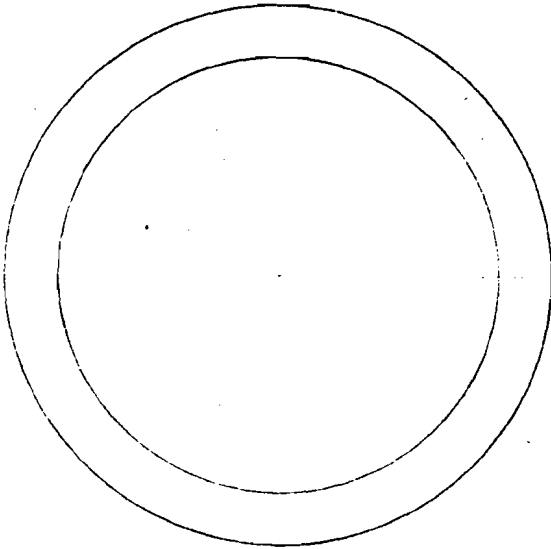
PART NO. 05.02.02.02
 MILD STEEL

16" DIA.



71

ENERGY ALTERNATION PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
MATCHER STEM AND PLATE (1 of 2)	
DATE 1 NOVEMBER 1975	05020201
DRAWN BY J. J. G.	



ITEM 3 P.M. RING
PART NO. 05.0202.03
MILD STEEL

NOM.
12-7/16"

10" NOM.



TOLERANCES UNLESS OTHERWISE SPECIFIED
 $\pm 1/16"$

SCALE: 1/2

6-3/32"

3-25/32"

2-7/32"



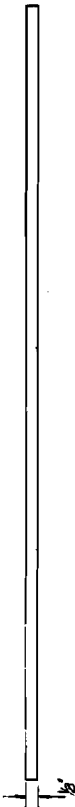
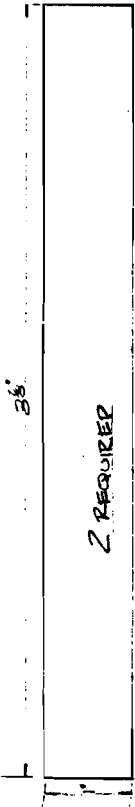
1-3/4"

3/16"



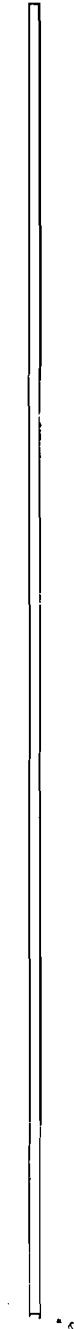
ITEM 4 P.M. RIB
PART NO. 05.0202.04
MILD STEEL
EIGHT PIECES REQUIRED

1/2"



OS020124 H

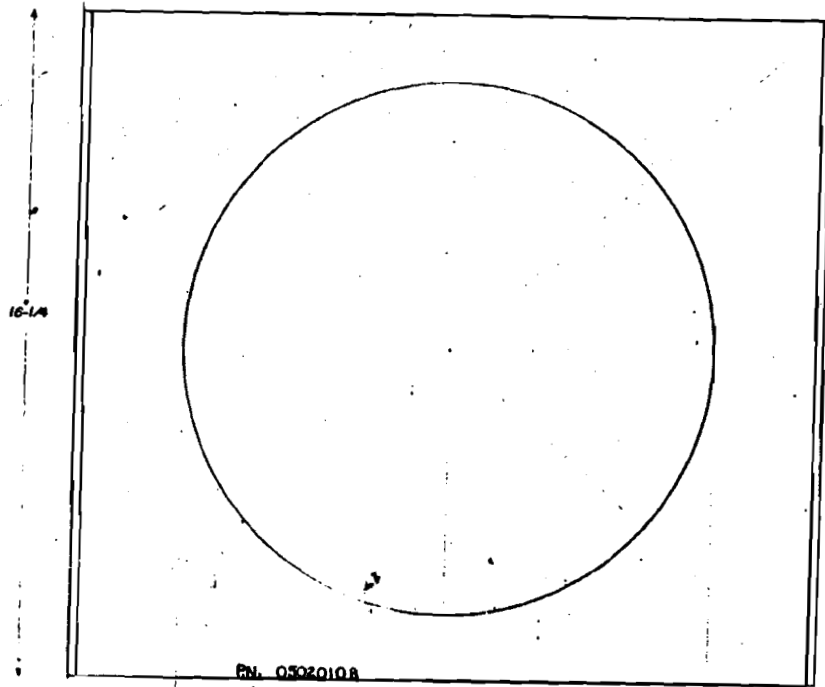
15'



OS020124 G

FRENCH ALTERNATIVE PROGRAM	
UNIVERSITY OF ALABAMA	
MISC. OPERATIONS	
SHT#	DATE BY ELOW
05020101	DATE - 6-1-76
	SCALE - FULL

18" NOM



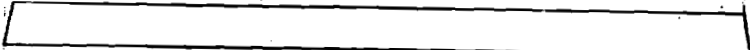
18-1/4"

P.N. 05020102

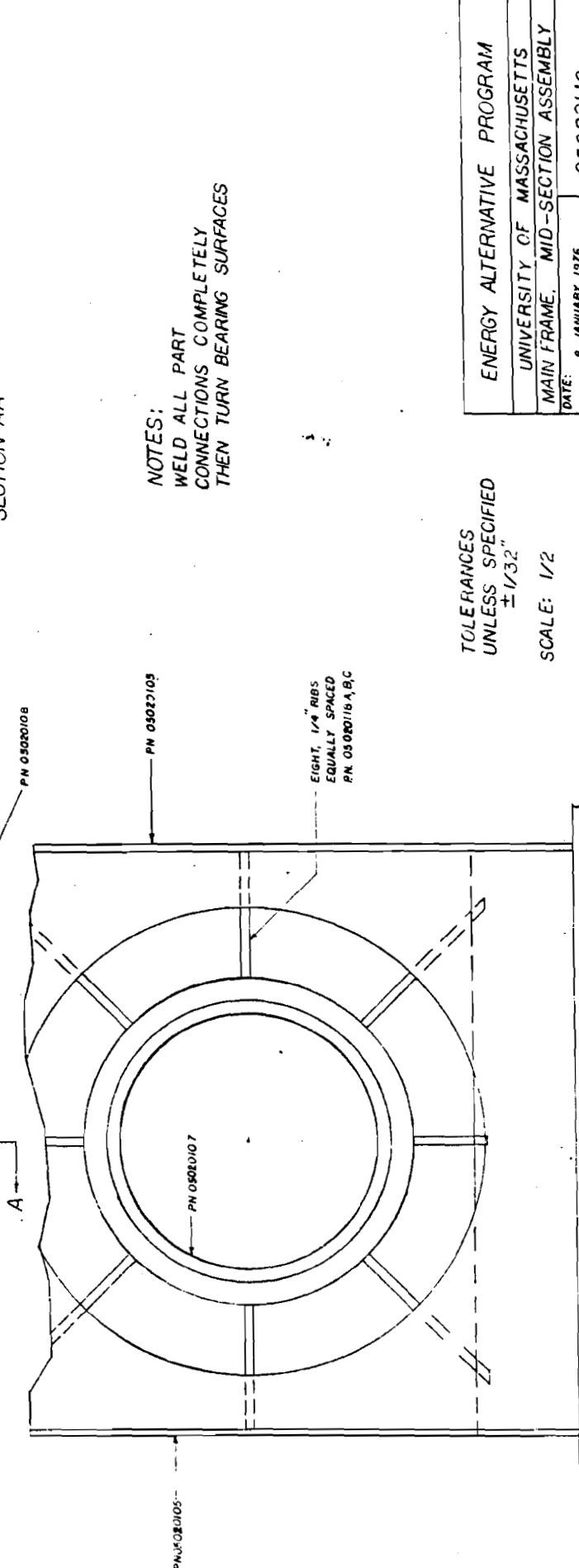
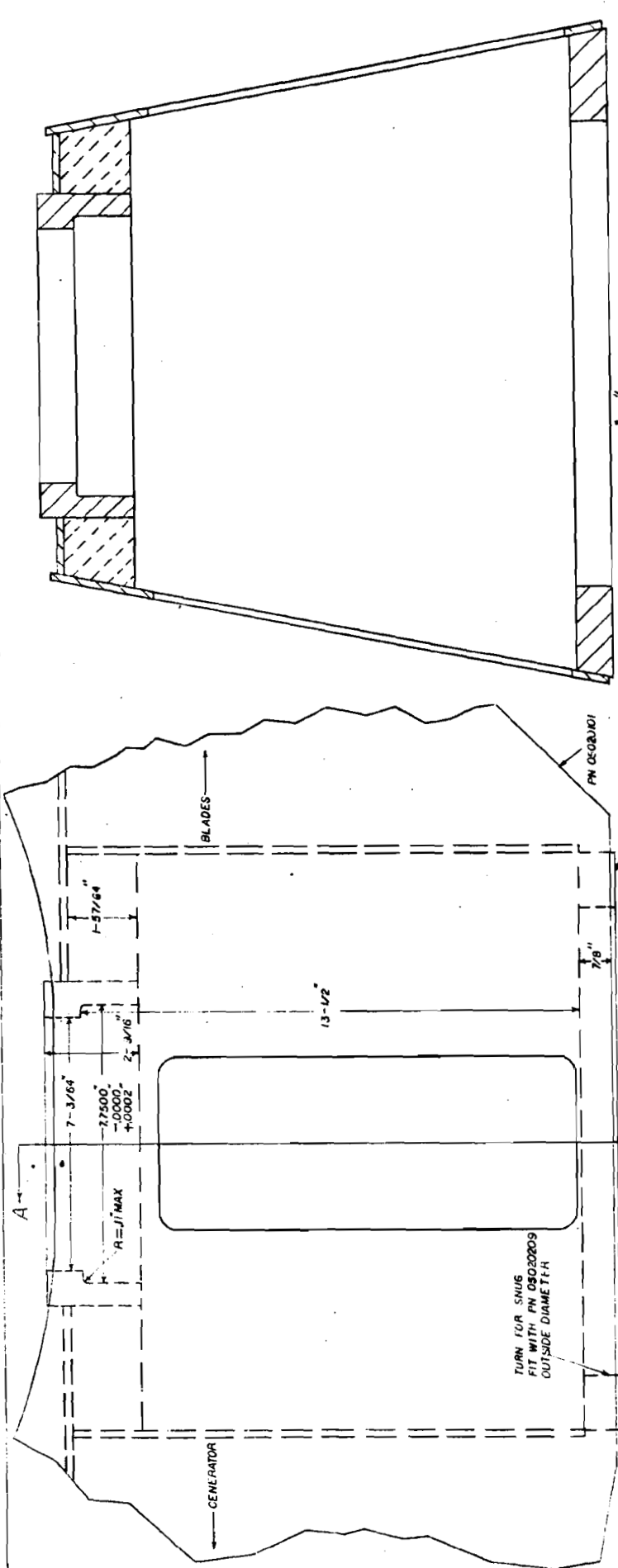
12-3/4"
NOM

NOTES:
LOCATE 12-3/4" CIRCLE BY CONSTRUCTING
DIAGONALS.
WHEN CUTTING THIS CIRCLE DO NOT DEVIATE
MORE THAN $\pm 1/4"$ AS THIS CIRCLE WILL BE
USED FOR FABRICATING P.N. 05020103.

CHAMFER ALONG EDGES
FOR WELDING WITH " "
P.N. 05020101



ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
PITCH BEARING-OUTER RACE	
DATE: 11 DECEMBER 1975	05020102
DRAWN BY: [signature]	



NOTES:
 WELD ALL PART
 CONNECTIONS COMPLETELY
 THEN TURN BEARING SURFACES

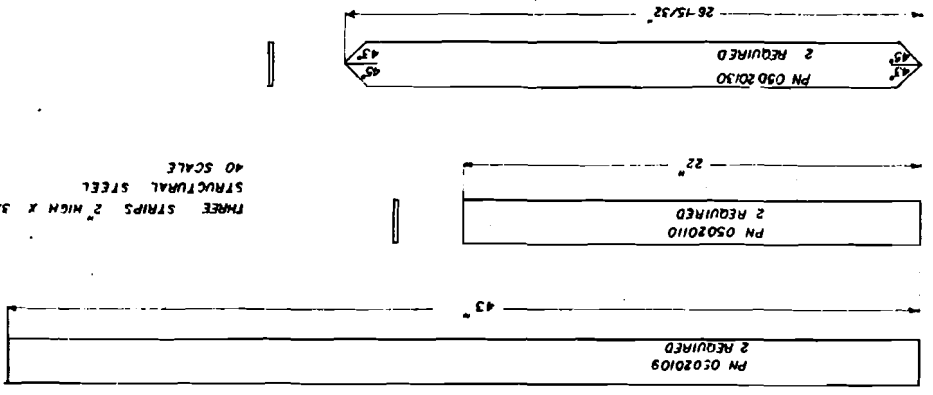
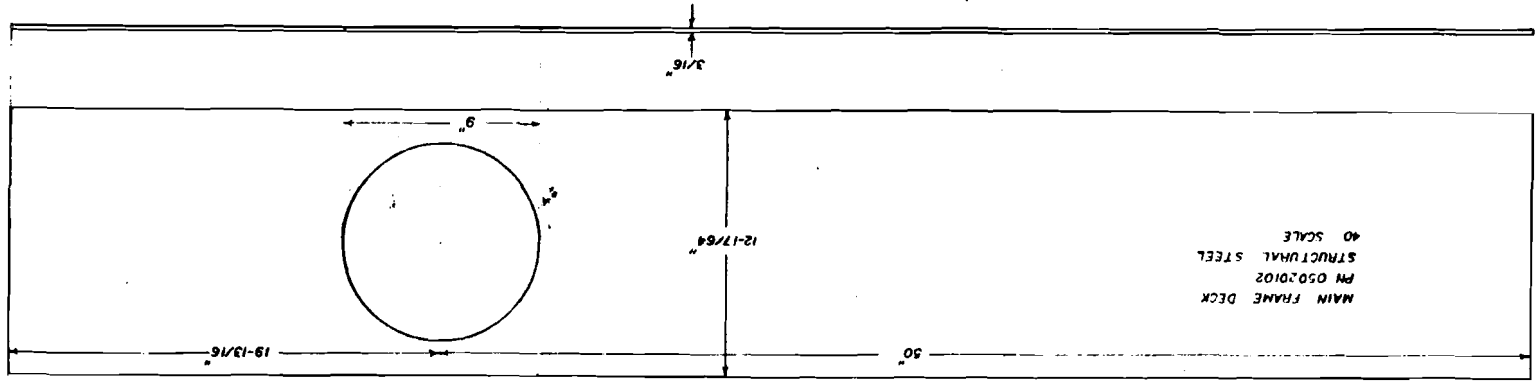
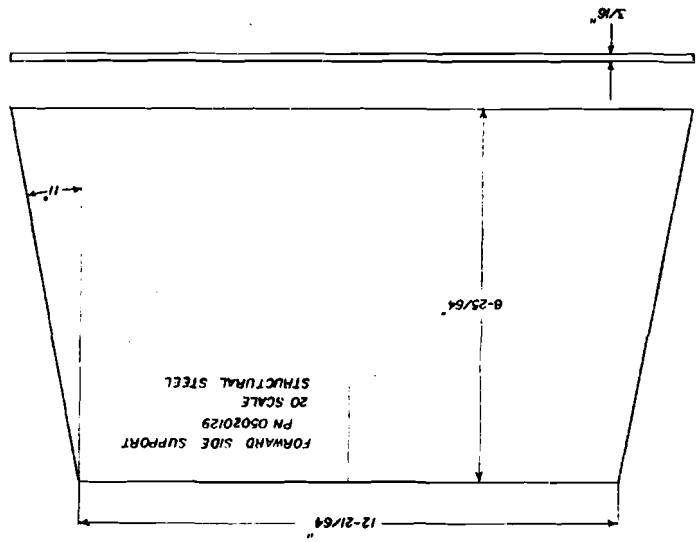
TOLERANCES
 UNLESS SPECIFIED
 ± 1/32"
 SCALE: 1/2

ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
MAIN FRAME, MID-SECTION ASSEMBLY	
DATE: 8 JANUARY 1976	05020110
DRAWN BY: M. G. S.	

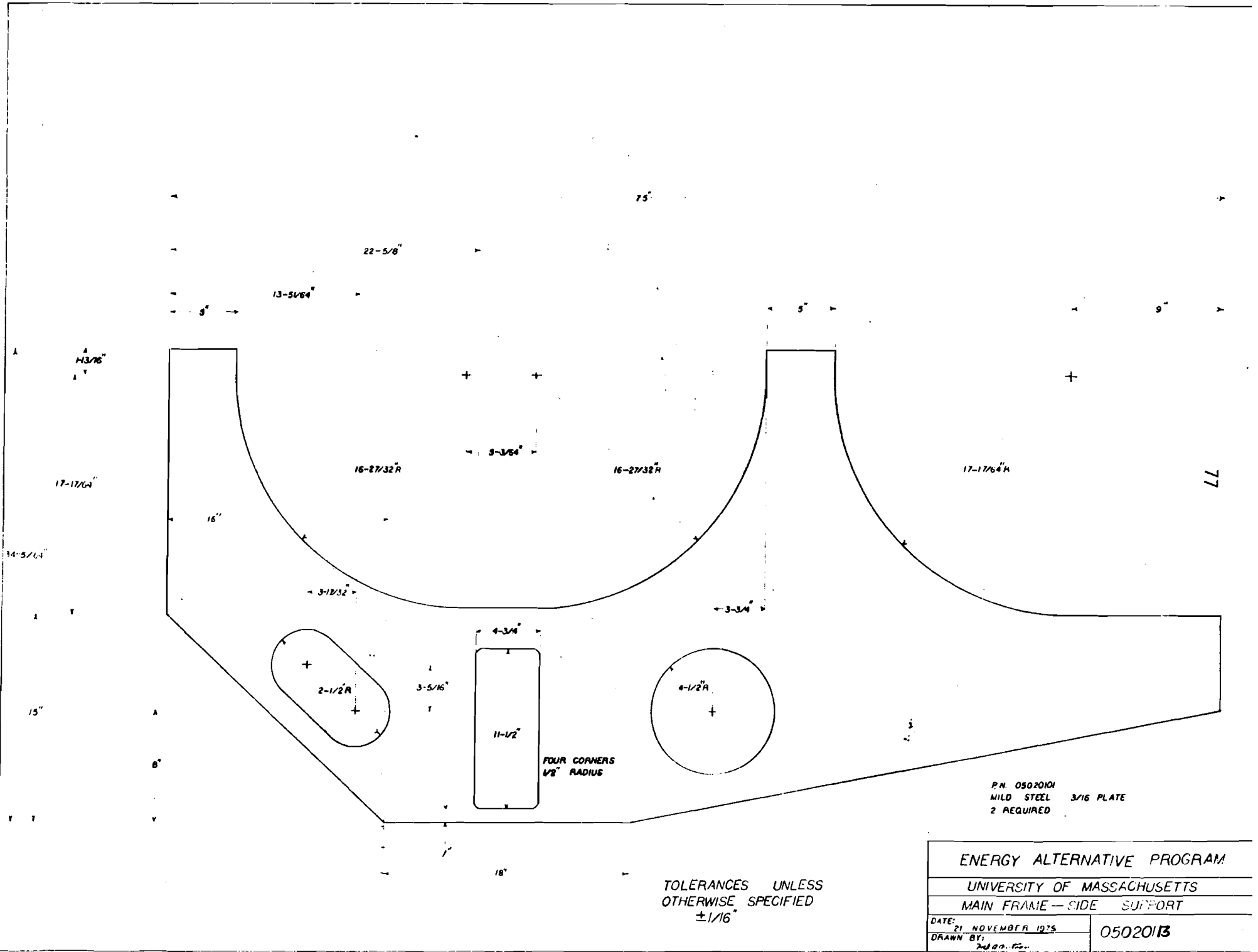
BOTTOM VIEW

ENERGY ALTERNATIVE PROGRAM
 UNIVERSITY OF MASSACHUSETTS
 MAIN FRAME - MISC. PARTS
 DATE 12 JANUARY 1971
 DRAWN BY
 05020112

TOLERANCES UNLESS SPECIFIED ±V32



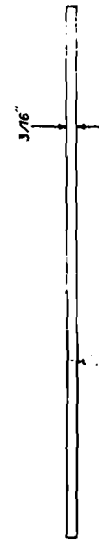
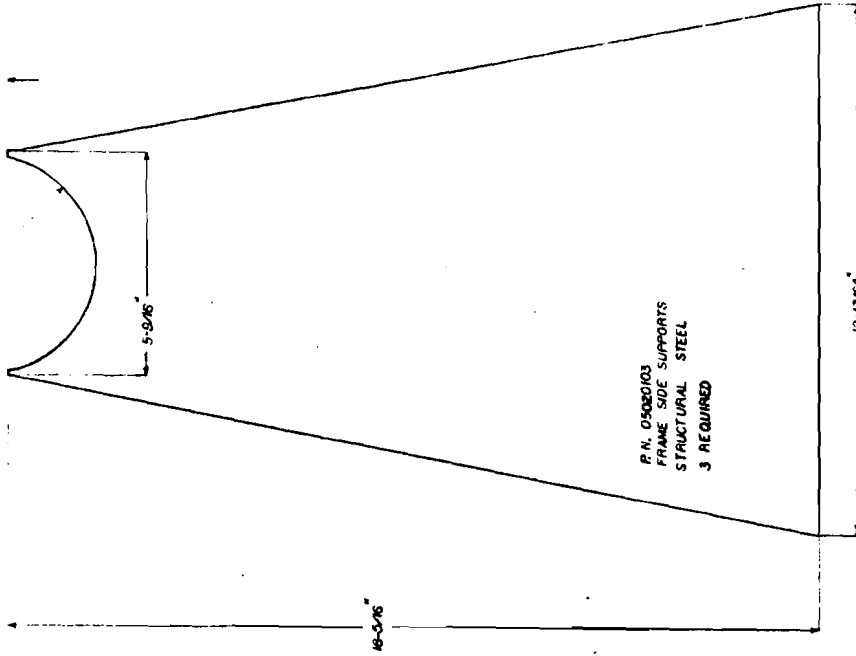
THREE STRIPS 2" HIGH X 3/16" THICK
 STRUCTURAL STEEL
 40 SCALE



TOLERANCES UNLESS OTHERWISE SPECIFIED $\pm 1/16"$

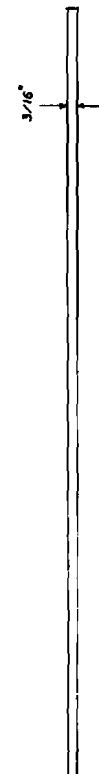
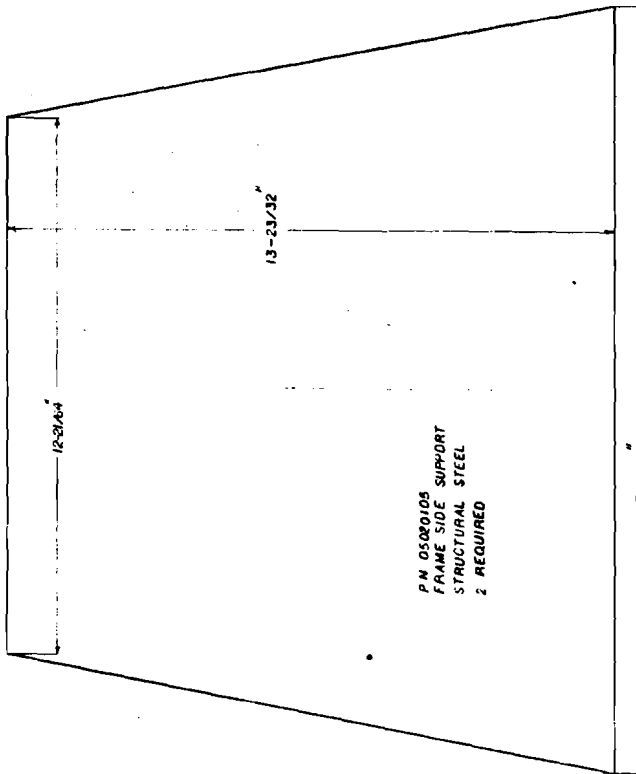
ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
MAIN FRAME - SIDE SUPPORT	
DATE: 21 NOVEMBER 1975	050201B
DRAWN BY: [Signature]	

2 1/2" RADIUS
1/4"



16-3/16"

12-1/16"



12-21/64"

13-23/32"

17-43/64"

TOLERANCES UNLESS OTHERWISE SPECIFIED
± 1/16"

SCALE: 1/2

ENERGY ALTERNATIVE PROGRAM

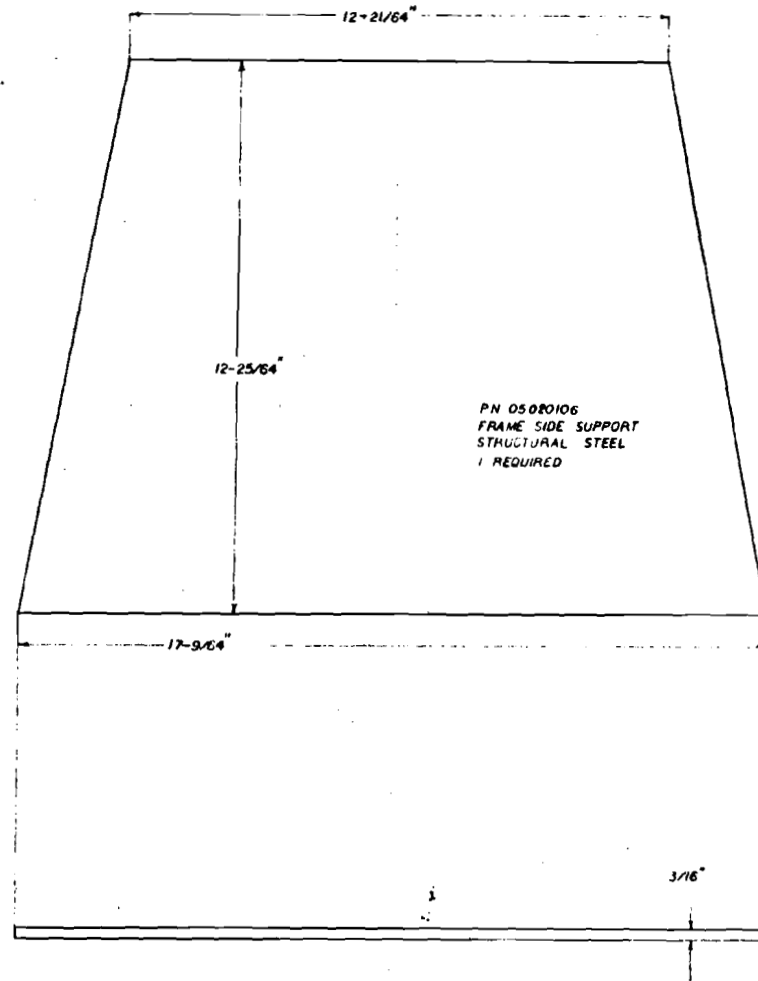
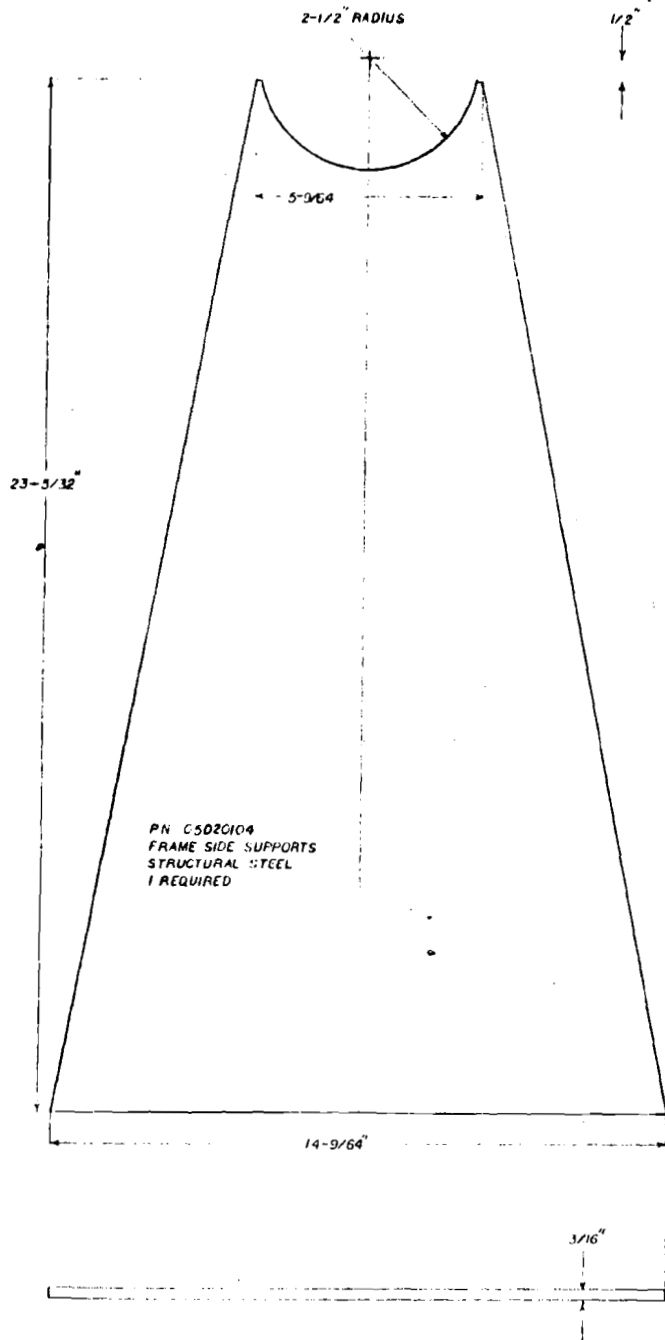
UNIVERSITY OF MASSACHUSETTS

MAIN FRAME SIDE SUPPORTS

DATE: 5 JANUARY 1976

DRAWN BY: J. G. G. G.

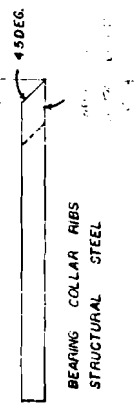
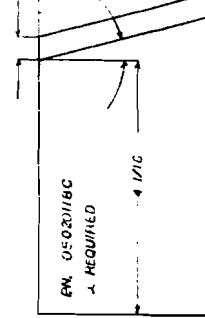
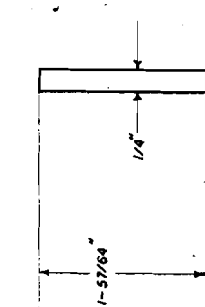
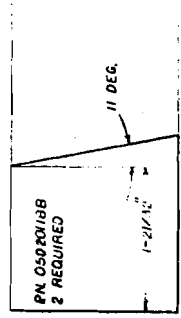
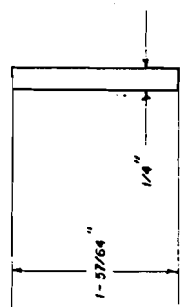
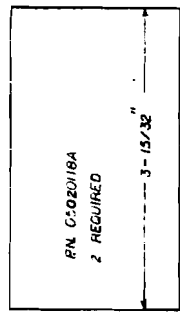
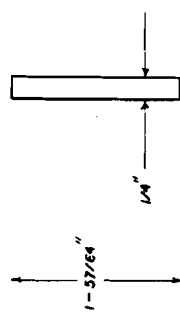
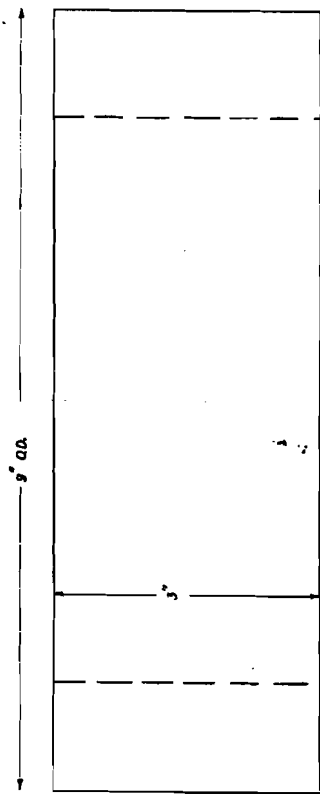
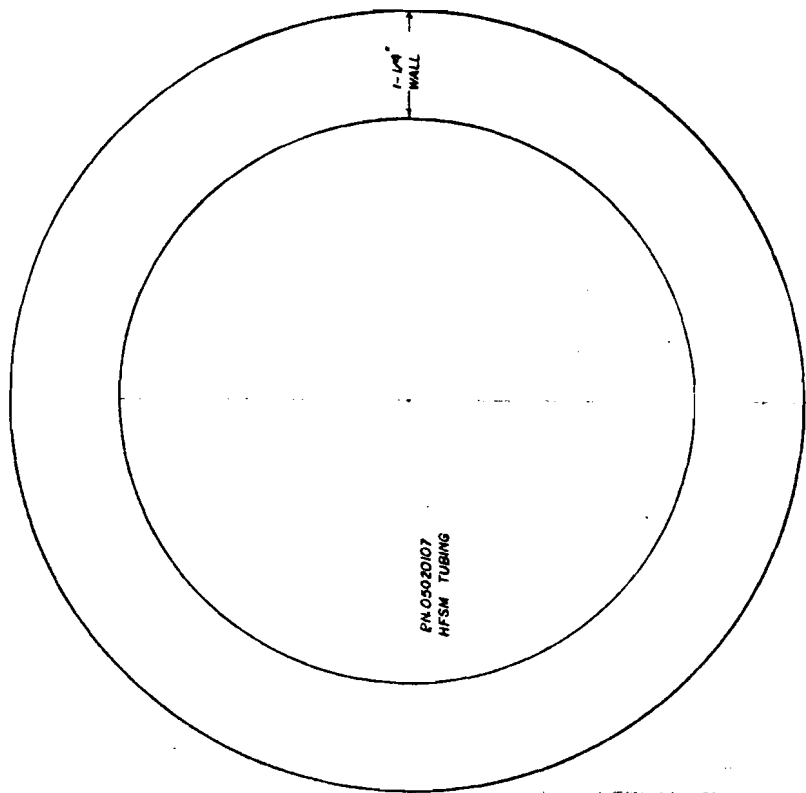
05020114



79

TOLERANCES UNLESS OTHERWISE SPECIFIED
± 1/16"
SCALE: 1/2"

ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
MAIN FRAME, SIDE SUPPORTS	
DATE: 1 JANUARY 1976	05020116
DRAWN BY: J.L.P. 12-76	



PN 05020107
HFSM TUBING

1-1/4\"/>

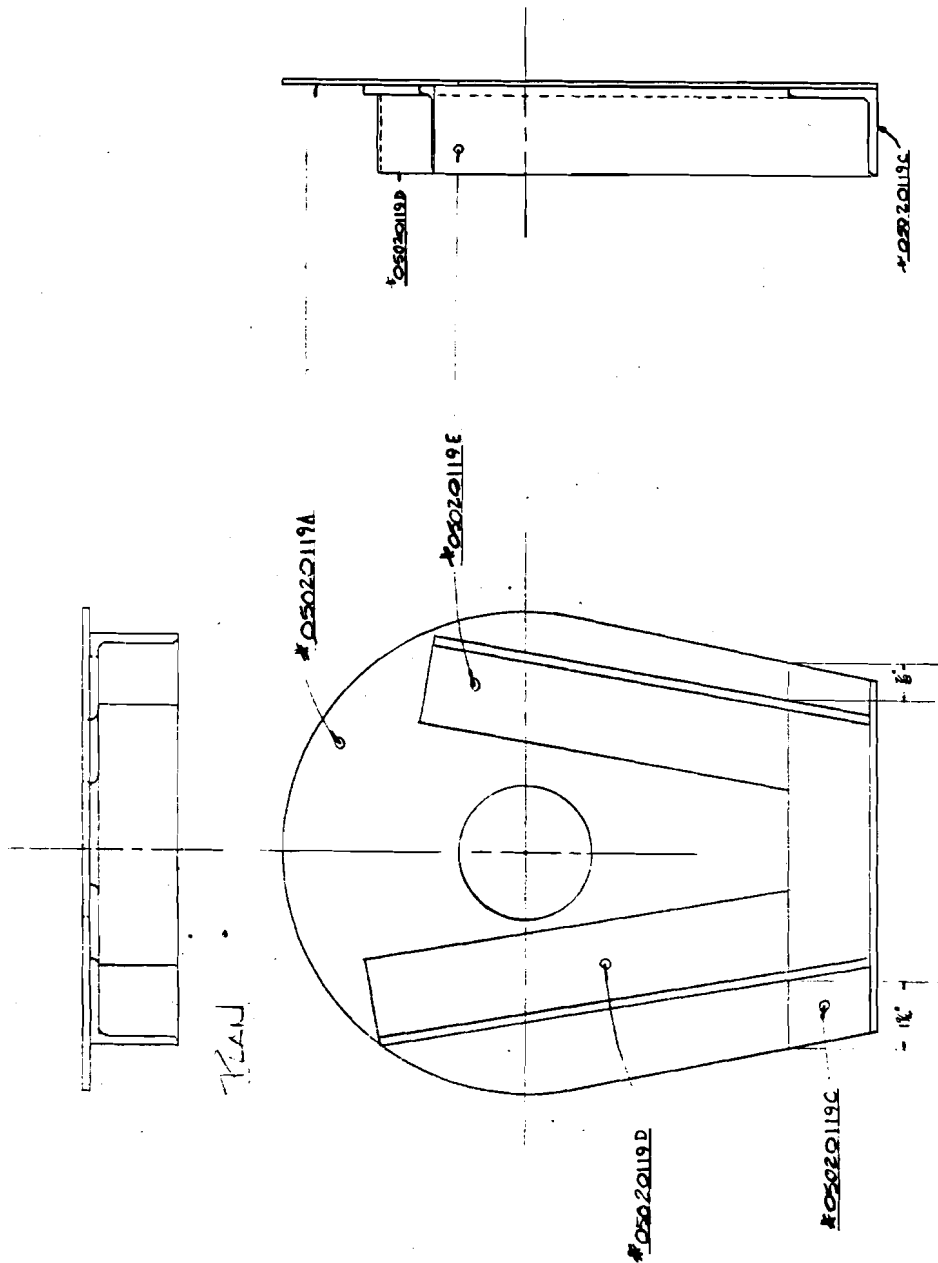
9\"/>

BEARING COLLAR RIBS
STRUCTURAL STEEL

TOLERANCES UNLESS
SPECIFIED $\pm 1/32$ "

FULL SCALE

ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
MAIN FRAME—BEARING COLLAR AND RIBS	
DATE:	10 JANUARY 1976
DRAWN BY:	J. G. G.
05020118	



R.T. SIDE

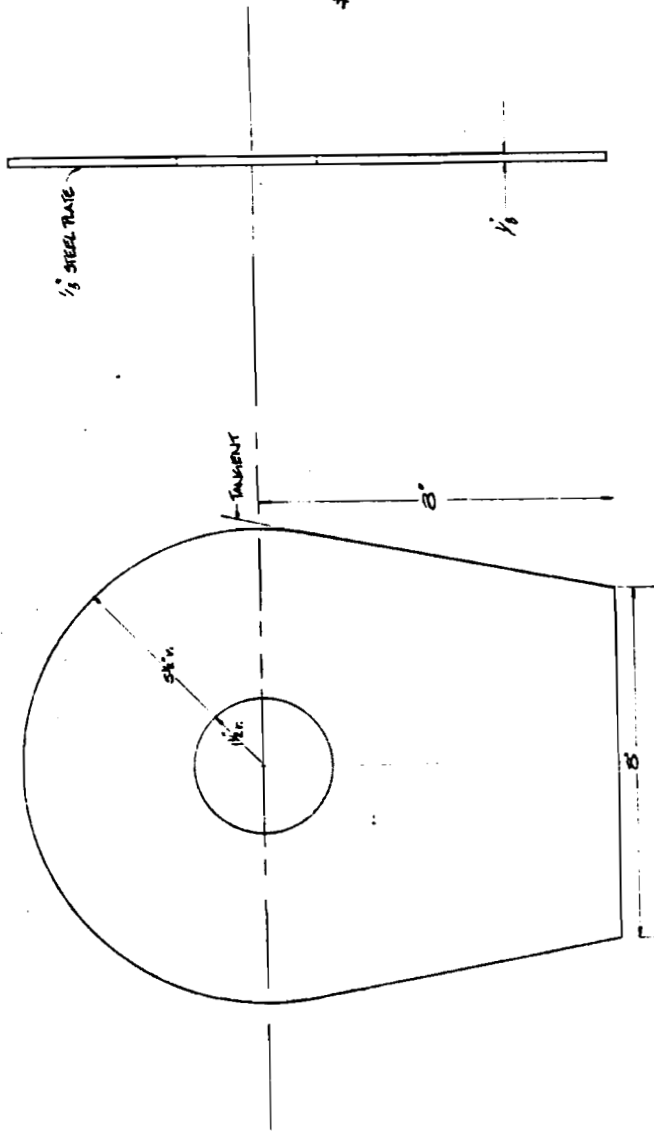
ELEVATION

PLAN

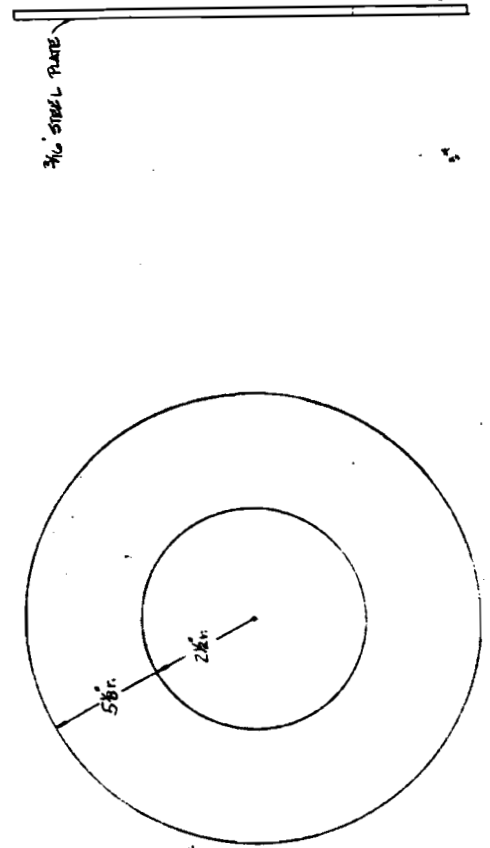
NOTE: ALL PARTS ARE WELDED

FRONT ELEVATION
SCALE 1/2" = 1"

ENERGY ASSISTANT'S PROGRAM	
UNIVERSITY OF CALIFORNIA	
PACIFIC COAST CAMPUS - ST. ANGELO, TX	
SHEET	PAVING BLOCKS
	DATE 5-20-76
DWG: 05020190	

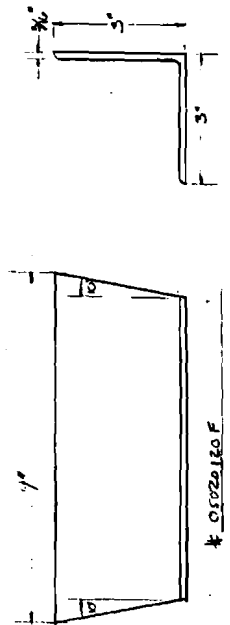
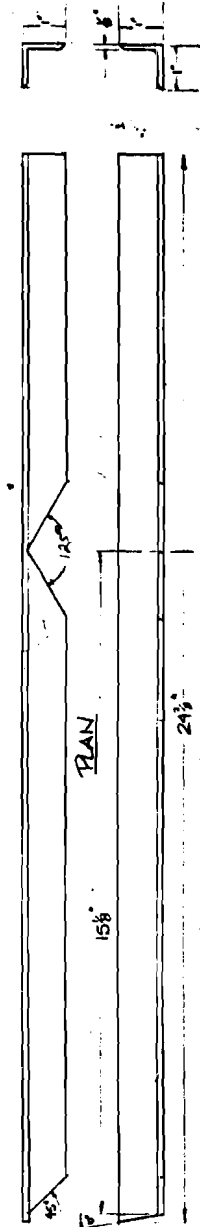
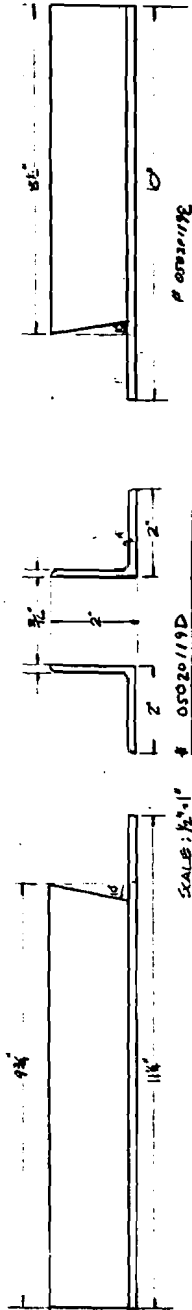
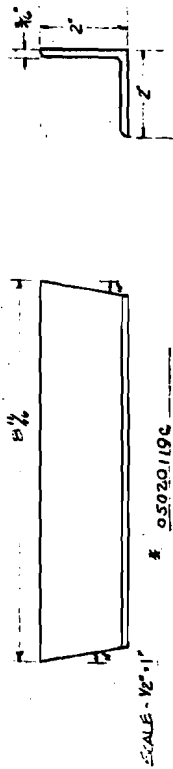
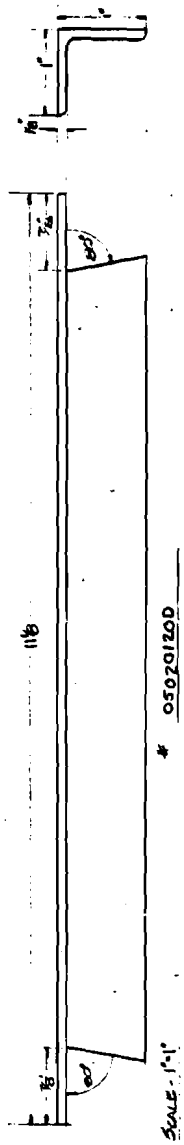
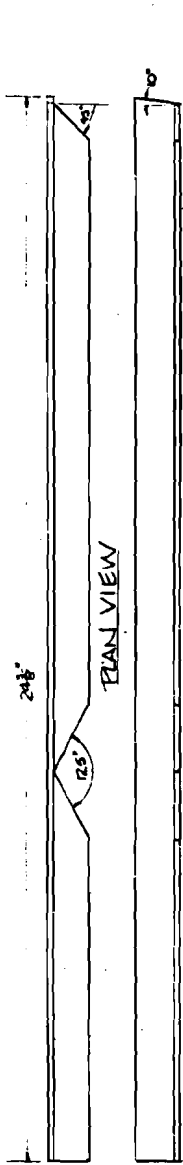


PINION BEARING BRACKET (PART NO. 050201194)
SCALE 1/2"=1"



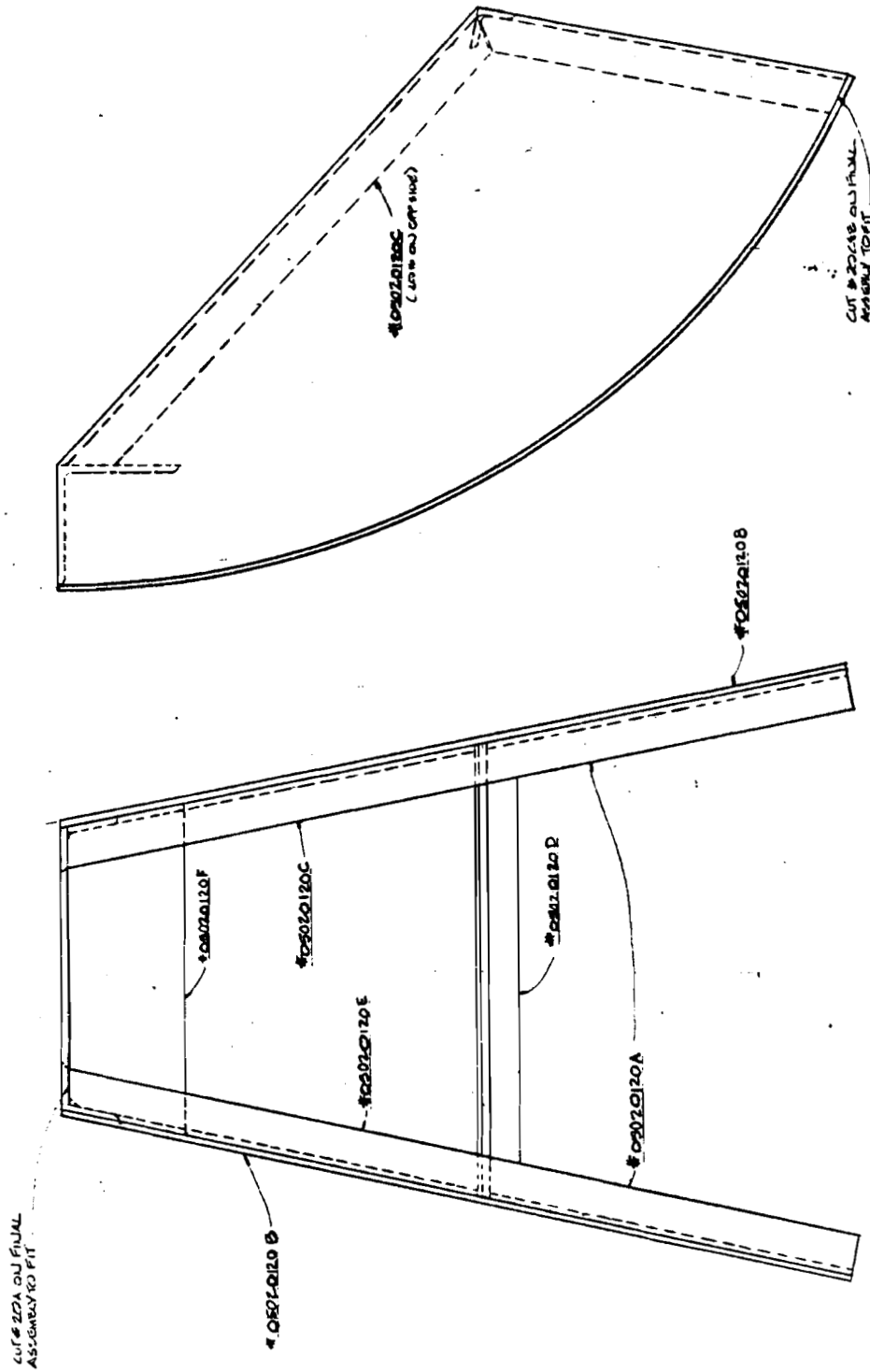
PINION BEARING SPACER (PART NO. 050201198)
SCALE 1/2"=1"

ENERGY ALTERNATIVE PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
TRAINING BRACKET MATH	
DATE BY: 1/11/82	DWG. NO.:
APP BY: 2/10/82	05020121



TOLERANCES UNLESS SPECIFIED.

ENERGY ALTERNATIVES PROGRAM	
UNIVERSITY OF MARYLAND	
UNION BEARING BEARINGS	DATE: 5-24-76
DESIGN NUMBER: 05020120	SCALE: AS SHOWN
DRAWN BY: [Signature]	

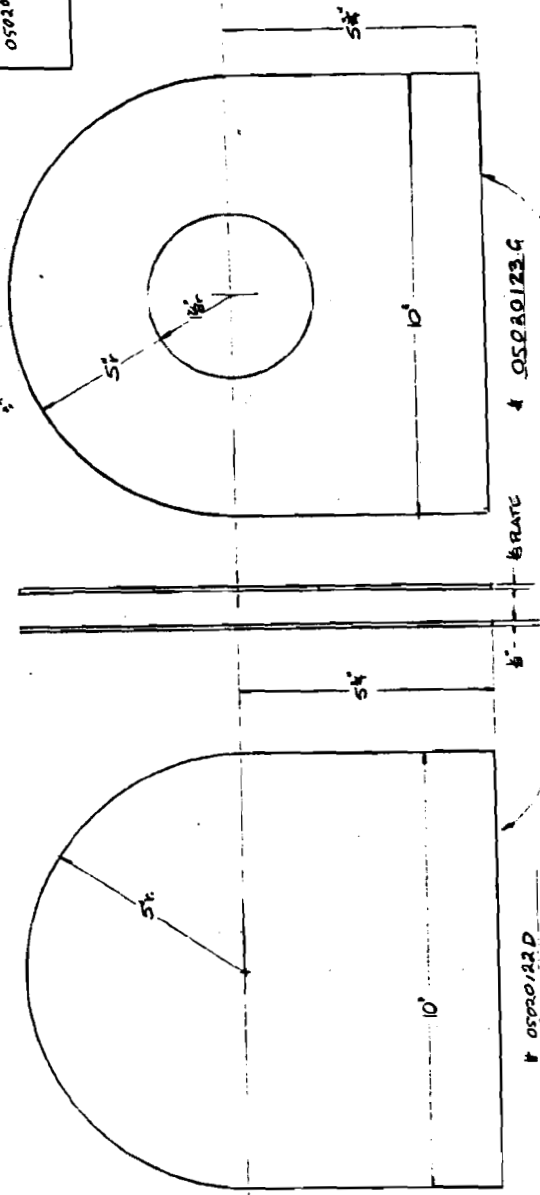


ENERGY ALTERNATIVES PROGRAM	
UNIVERSITY OF MARYLAND	
TRINITY PARK'S FOUNDATIONAL A-100	
SAT #	TRINITY PARKS
	DATE: 5-22-76
	APP:
TRAF #05020123	SCALE 1/2"=1"

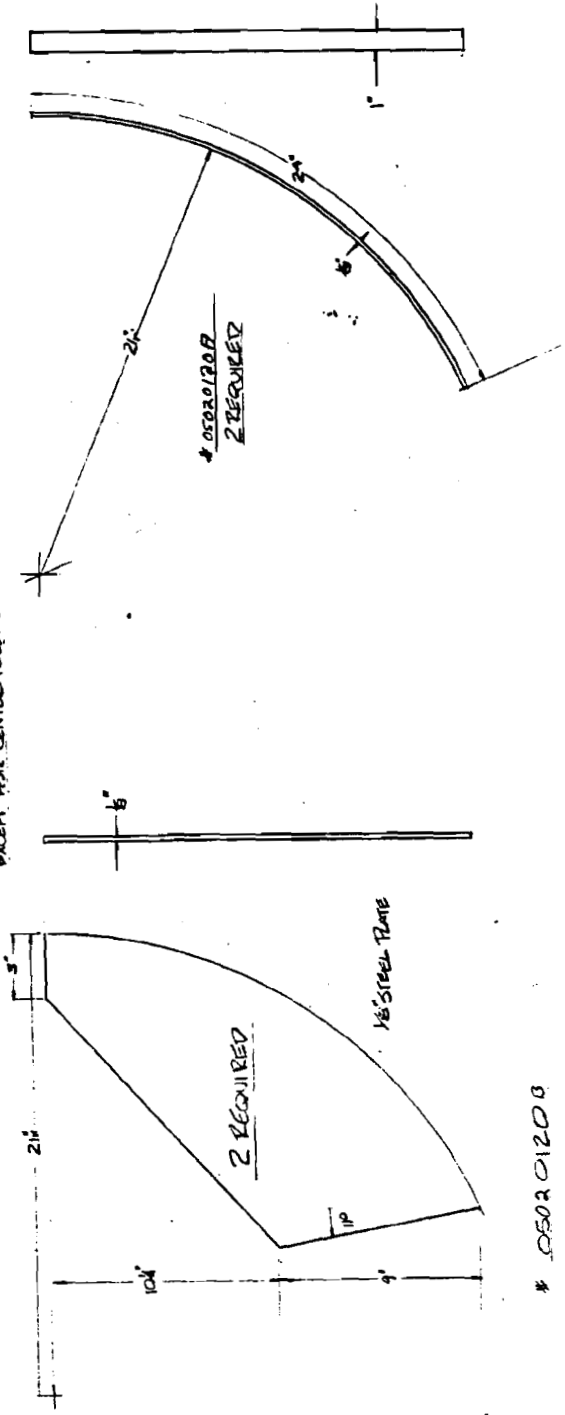
TRINITY PARKS FOUNDATION
SCALE 1/2"=1"

201000.50 a mark

ENERGY ALTERNATIVES REGIONAL
UNIVERSITY OF MASSACHUSETTS
TRUCKS BEARING, NISC.
SHEET # 0502014
DESIGN BILLING SCALE: AS SHOWN APP. DATE: 5-29-76



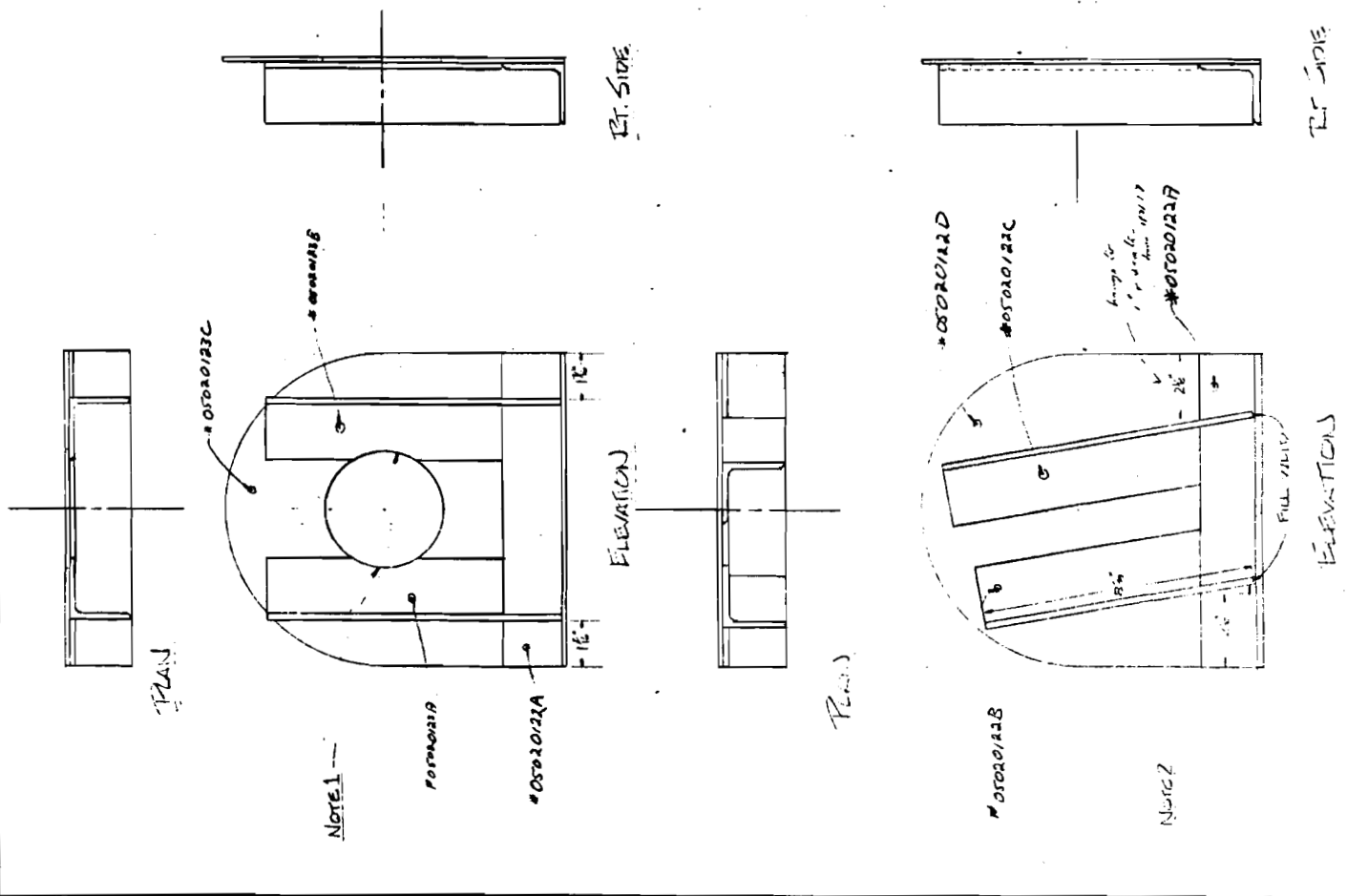
NOTE: THESE TWO BEARS ARE IDENTICAL EXCEPT FOR CENTER HOLES.

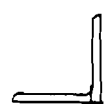
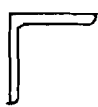
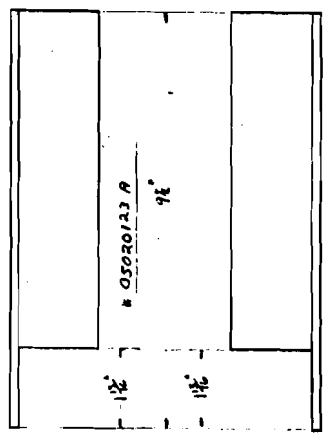
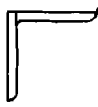
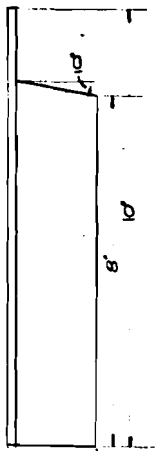
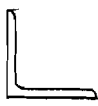
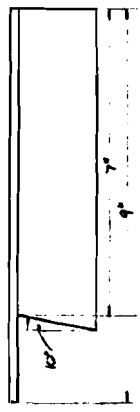
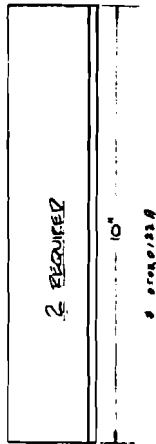
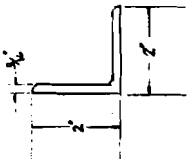


05020120 B
TRUCK BEARING FOUNDATION
SCALE: 1/4"=1"

ENERGY ALTERNATIVES KITCHEN	
UNIVERSITY OF MASSACHUSETTS	
GENERATION TECHNOLOGY PROJECT ASSEMBLY	
PART #	OS2020130
DATE	5-29-76
APP.	JKM
SCALE	3/8" = 1"

- NOTES:
1. FILE RECES OS2020130 TO DIAMETER OF CENTER HOLE OR OS2020131 ON FINAL ASSEMBLY.
 2. BEFORE ASSEMBLY REAM PIPES OS2020128 TO OVERALL LENGTH OF 8 3/8"
- * ALL JOINTS ARE TO BE WELDED

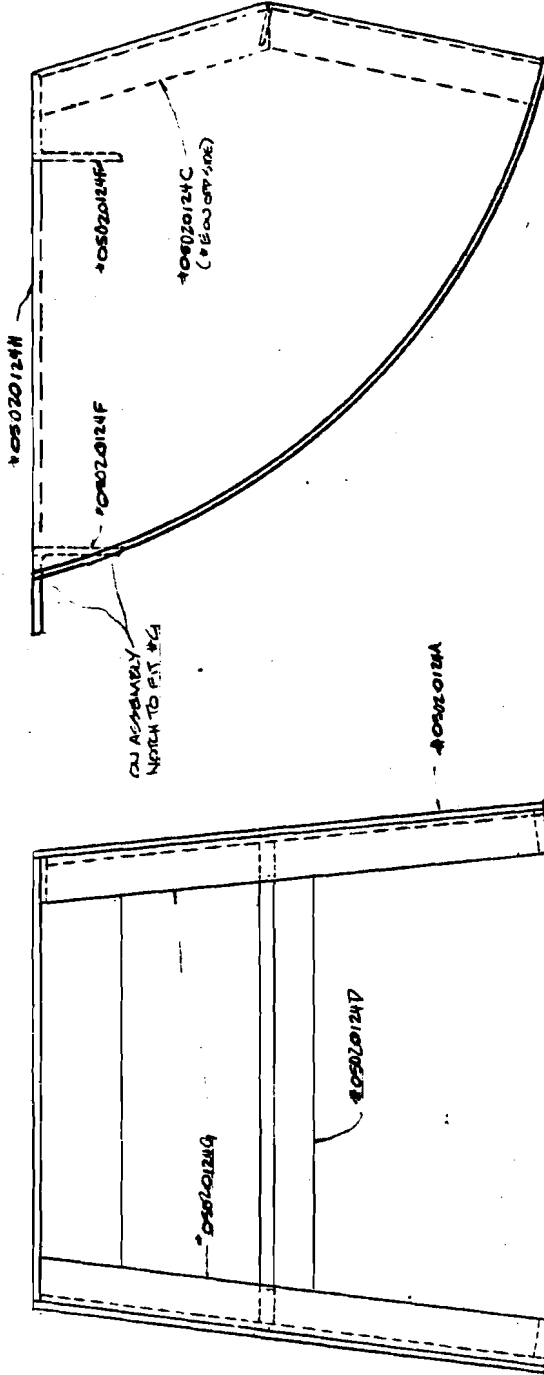




NOTE:
ALL PIECES CUT FROM 2x2x3/8 ANGLE STK.
TOLERANCE 1/8" UNLESS SPECIFIED

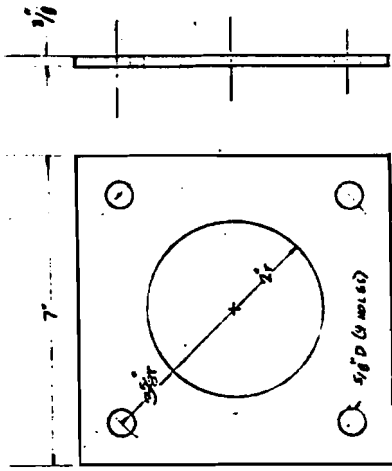
ENERGY ALTERNATIVES PROGRAM	
UNIVERSITY OF MASSACHUSETTS	
05020131	
DATE:	SCALE: 1/2"=1'
DRAWN BY:	DATE: 11-7-76
APP. BY:	

ALL JOINTS WELDED



GENERATOR BEARING FOUNDATION ASSEMBLY
SCALE 1/2"=1"

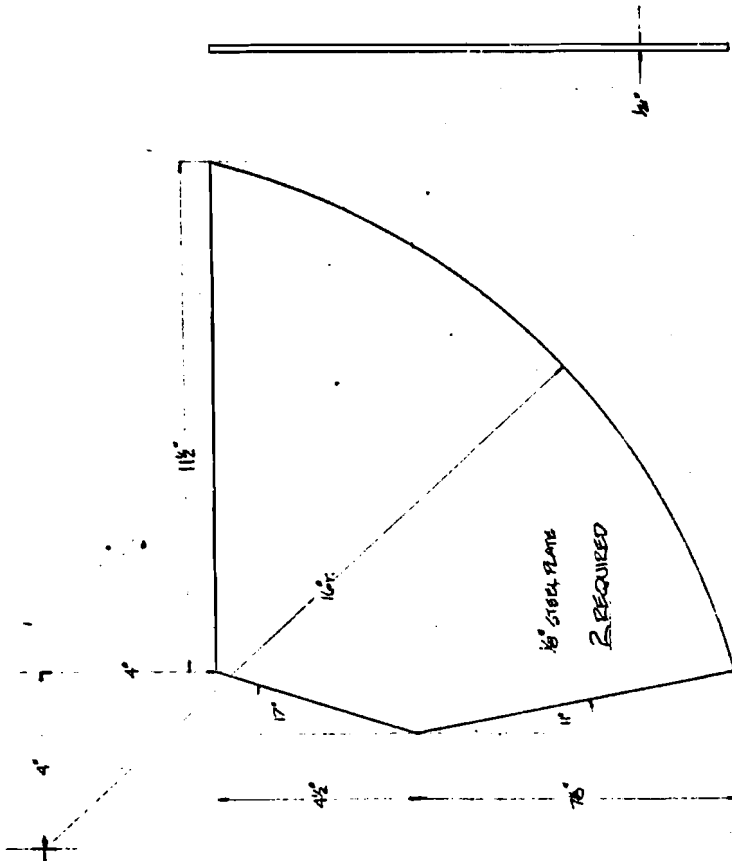
ENERGY ALTERNATIVES PROGRAM	
UNIVERSITY OF MICHIGAN	
CIENERGY/ALTERNATIVE FOUNDATION ASS.	
WT #	TRUMP BLDG
	DATE 5-25-76
	APP
DWG # 05020124	SCALE 1/2"=1"



OS020129 E

ROUND ALL CORNERS TO R1/4"

GENERATOR BEARING SPACER
SCALE 1/2"=1'



GENERATOR FOUNDATION SIDES
SCALE 1/2"=1'

OS020129 A

TOLERANCE 1/8" UNLESS SPECIFIED

ENERGY ALTERNATIVES PROGRAM	
UNIVERSITY OF WISCONSIN	GENERATOR FOUNDATION SIDE
PROJECT NO. 70	DATE: 8/1/80
APP. 7/10/80	SCALE: AS SHOWN
05020133	

ENERGY ALTERNATIVES PROGRAM

UNIVERSITY OF MASSACHUSETTS

GENERATOR FOUNDATION MISC.

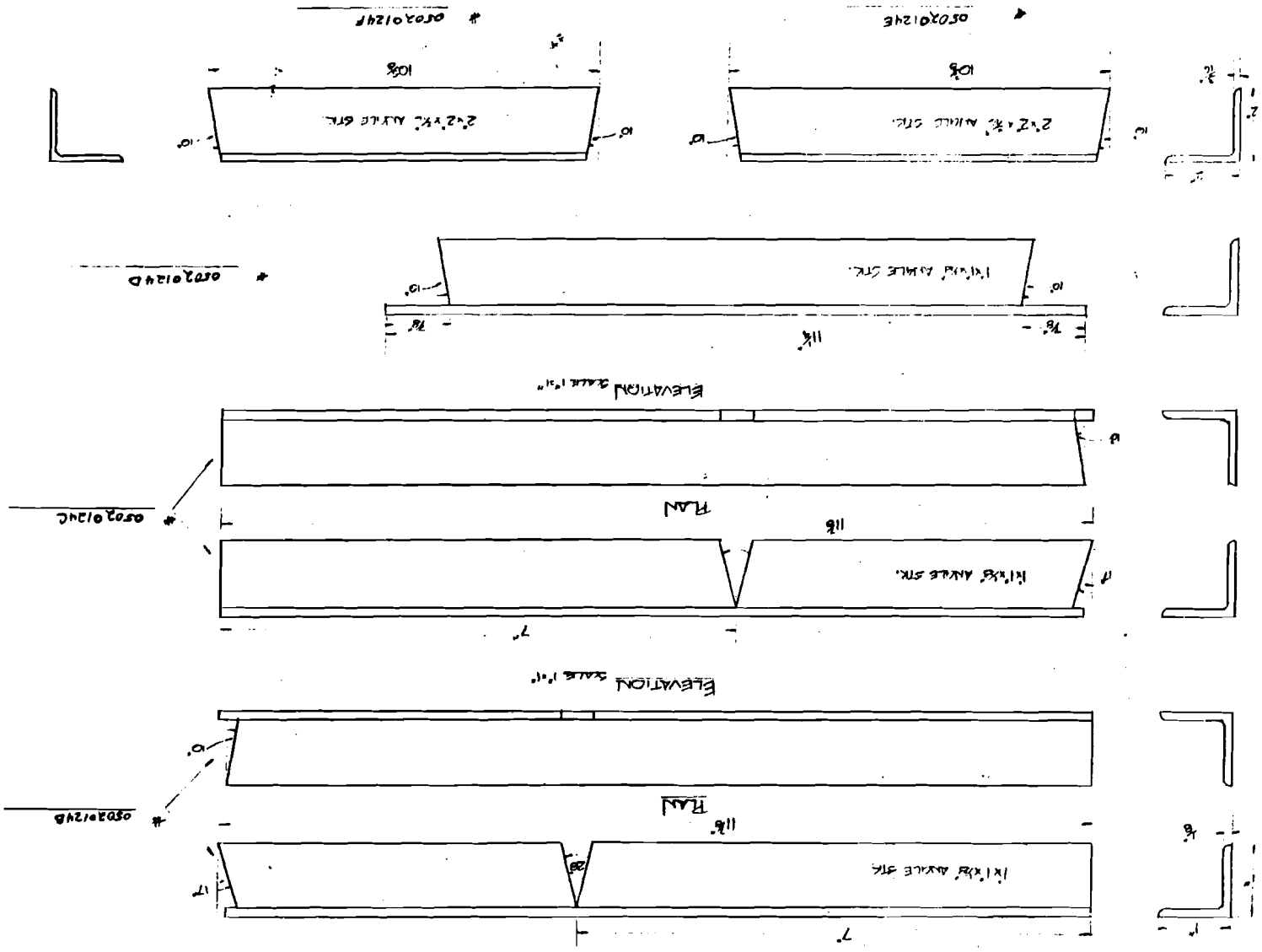
DRAWN BY: J. KIM
DATE: 5-24-76

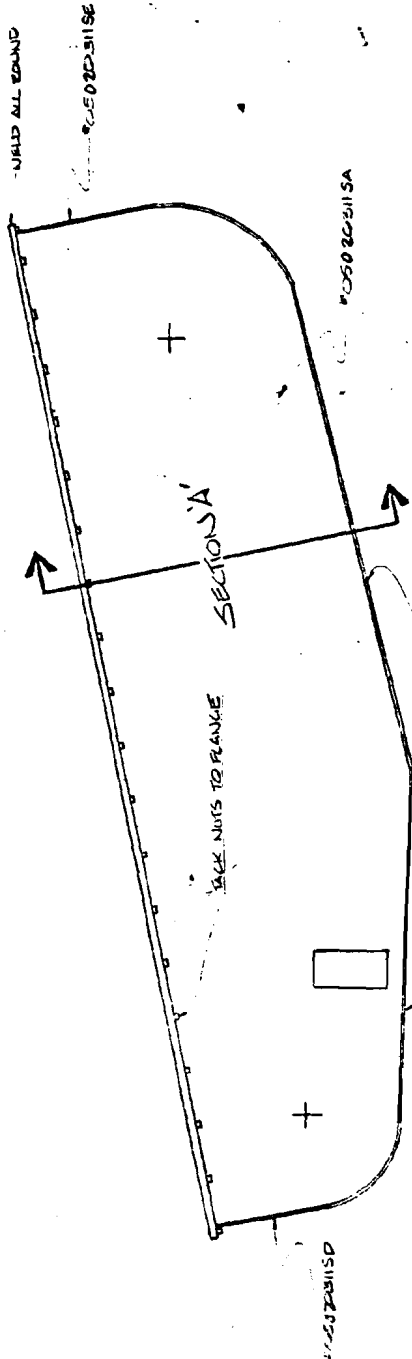
DRAWING # 05020124

SCALE AS SHOWN

GENERATOR FOUNDATION MISC.

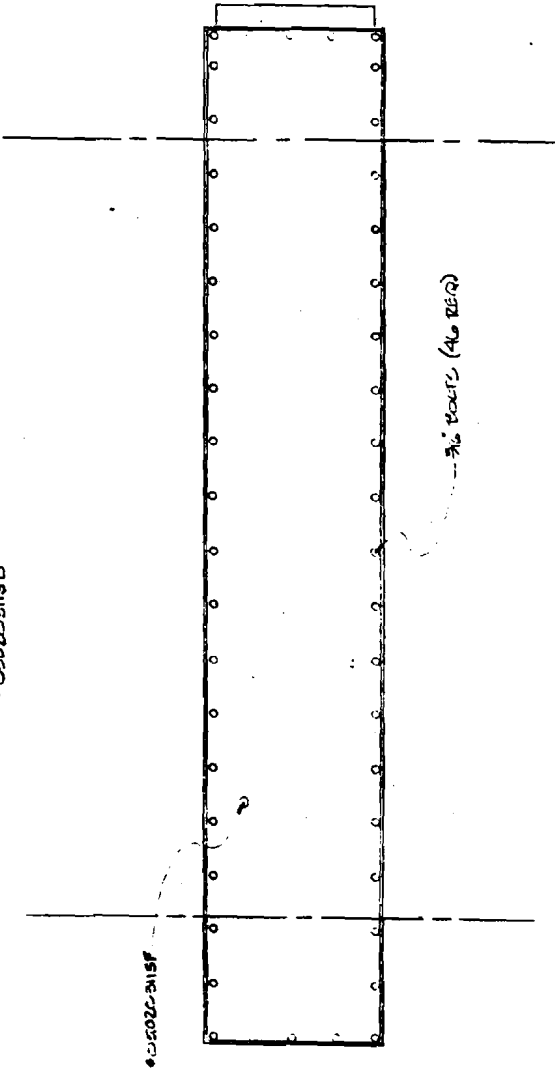
RELEASE TO UNLESS OTHERWISE SPECIFIED





SECTION A

WELD ALL ROUND
(A502C-315B)

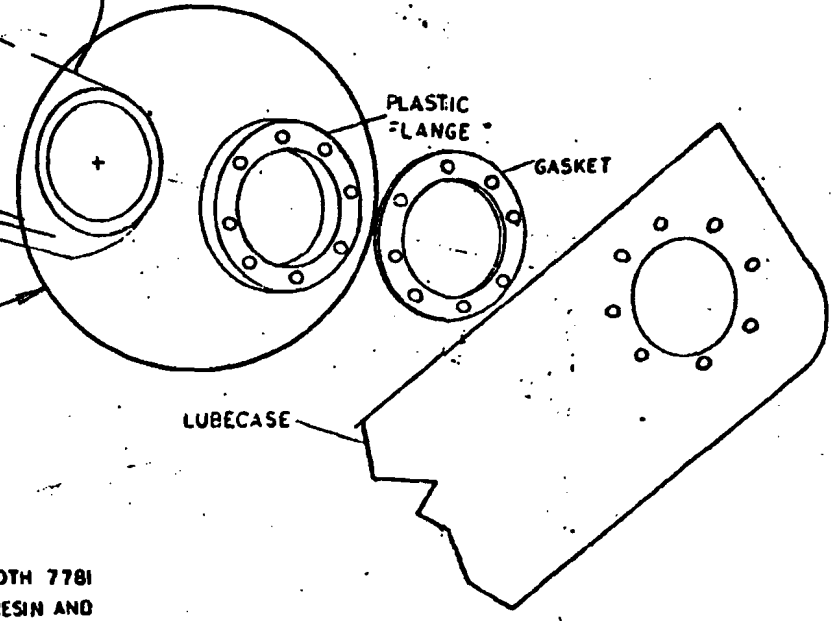
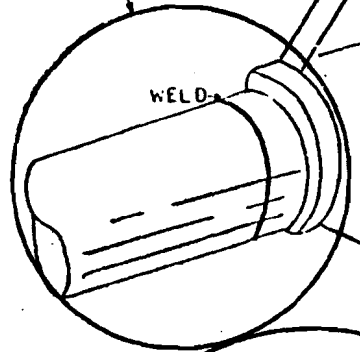
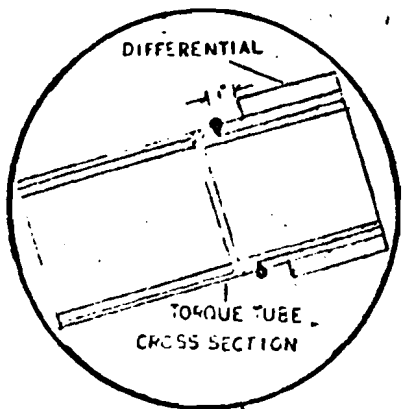


STEEL LUBRICATION CASE ASSEM.

SCALE 3/8" = 1"

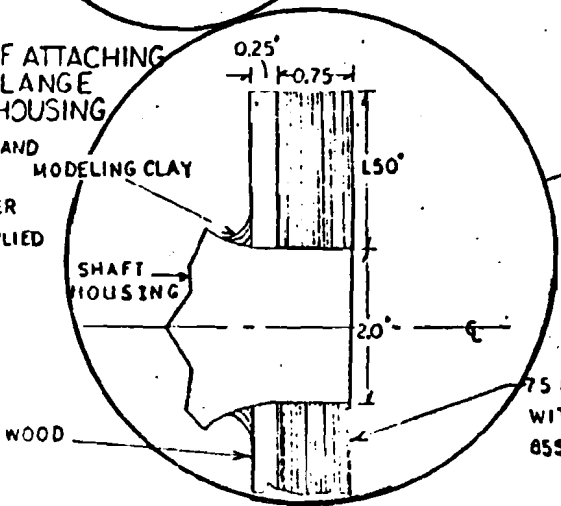
ENGINEERING/DESIGN/TEST/CONSTRUCTION	
PROJECT NO. 05020310	
SHEET #	05020310
DESIGNED BY	DAVEY T. LINDSAY
DATE	7/20/76
APP.	
SCALE	3/8" = 1"

ENERGY ALTERNATIVE PROGRAM
 U-MASS
 REAR AXLE ASSEMBLY
 HOUSING MODIFICATIONS
 DRWNG NO. 05030111
 DRN BY: PAUL DONOHUE
 DATE: 8-2-77



METHOD OF ATTACHING
 PLASTIC FLANGE
 TO SHAFT HOUSING

NOTE: WOOD AND
 CLAY ARE
 REMOVED AFTER
 CLOTH IS APPLIED
 AND CURED



75 PLYS OF GLASS CLOTH 7781
 WITH 7818 EPOXY RESIN AND
 855 EPOXY CURING AGENT

NOMENCLATURE

ID	inside diameter
OD	outside diameter
Kw	kilowatts
rpm	revolutions per minute
mph	miles per hour
V_o	free windstream velocity
ρ	air clevity
A	swept area of a wind rotor
Ω	rotational speed of a wind rotor
R	wind rotor blade radius
S-F	safety factor
A_p	cross sectional area of a steel pipe
#	pounds force
psi	pounds per square inch
WECS	Wind Energy Conversion System

BIBLIOGRAPHY

1. Baumeiste, Theodore, Marks' Standard Handbook for Mechanical Engineers, McGraw-Hill, New York, 1971.
2. Butterfield, Charles B., Jr., "Design of Hub and Pitching System for Wind Furnace I (WF-1)", UMass/Amherst Energy Alternatives Program, WF/TR/77/4.
3. Heronemus, William E., "Investigation of the Feasibility of Using Windpower for Space Heating in Colder Climates", UMass/Amherst Energy Alternatives Program, WF/PR/77/1.
4. Mabie, H. H. and F. W. Ocvirk, Mechanisms and Dynamics of Machinery, John Wiley and Sons, New York, NY, 1975.
5. "Morse Power Transmission Products Catalogue", Morse Chain, Division of Borg Warner Corp., 1973.
6. "Shot-Peening Applications", Metal Improvement Company Publication, Fourth edition.
7. Spotts, M. F., Design of Machine Elements, Fourth edition, Prentice Hall, Englewood Cliffs, NJ, 1971.
8. Stoddard, F. S., "Discussion of Momentum Theory for Windmills", UMass/Amherst Energy Alternatives Program, TR/76/2.